

THE DESIGN-STUDY OF A 10 H. P. HIGH DUTY
STATIONARY GAS ENGINE

BY

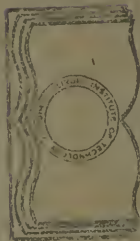
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ARMOUR INSTITUTE OF TECHNOLOGY

1917

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Plocinsky, A. J.

The design-study of a 10

H.P., high-duty, stationary

THE DESIGN-STUDY OF A 10 H.P., HIGH-DUTY, STATIONARY GAS ENGINE

A THESIS

PRESENTED BY

ABE J. PLOCINSKY AND CHAS. R. POMEROY

TO THE

PRESIDENT AND FACULTY

OF

ARMOUR INSTITUTE OF TECHNOLOGY

FOR THE DEGREE OF

BACHELOR OF SCIENCE

IN

MECHANICAL ENGINEERING

MAY 31, 1917

APPROVED:



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Dean of Engineering Studies



Dean of Cultural Studies

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FOREWORD.

In the presentation of this thesis, it is hoped that the present work will form the basis for the continued study of the design and the ultimate construction and testing of this proposed engine.

The writers wish to express their appreciation of the kind assistance of Professor Daniel Roesch.

A.J.P.

C.R.P.

Table of Contents.

	Page.
Foreword-----	1
Table of Contents-----	2
List of Illustrations-----	4
List of Symbols-----	6
Object-----	9
Progress Thesis-----	9
Scope of Present Work-----	9
Method-----	10
I. Preliminary Considerations-----	10
II. Preliminary Calculations-----	12
(a) Rated I.H.P.-----	12
(b) Max. B.H.P. and Max. I.H.P.-----	12
(c) Volume Ratio-----	14
(d) Percent of Clearance-----	16
(e) Bore and Stroke-----	16
(f) Clearance-----	18
(g) Actual R.P.M.-----	18
III. Cylinder-----	19
IV. Frame-----	27

REIGN OF

The first year of the reign of King Henry the Fifth was spent in the preparation of the army for the expedition to France. The king, who was then only twenty years of age, was determined to assert his claim to the French throne, and to avenge the death of his father, King Henry the Fourth. He had been educated in the most liberal manner, and was well versed in the arts of war and peace. He was also a great lover of his country, and was determined to defend its liberties against any foreign invasion. The king's army was composed of the best soldiers of the kingdom, and was well equipped with arms and armor. The king himself led the army in person, and was distinguished by his valor and skill in the field. The first battle of the reign was the battle of Agincourt, which was fought on the fifth day of the month of September, in the year of our Lord one thousand four hundred and fifteen. The king's army was victorious, and the French king, King Charles the Sixth, was forced to flee the field. This battle was one of the most famous in the history of England, and it established the reputation of King Henry the Fifth as a great warrior and a great leader. The king's army then marched on to Paris, and the king entered the city in triumph. The king's reign was a period of great glory and success for England, and it was during this time that the kingdom reached the height of its power and influence. The king's death, which occurred on the eighth day of the month of August, in the year of our Lord one thousand four hundred and fifty-eight, was a great loss to the kingdom, and it was followed by a period of civil war and confusion. The reign of King Henry the Fifth was a period of great importance in the history of England, and it was during this time that the kingdom was established as a great power in Europe.

Table of Contents, Continued.

	Page.
V. Valve Gear-----	35
VI. Flywheel and Crankshaft-----	44
VII. Reciprocating Parts-----	47
VIII. Means Provided for Experiment-----	55
Conclusion-----	60
Future Considerations-----	60
Appendix-----	62
Bibliography-----	62

List of Illustrations.

Sketches	Page.
Fig.1. Volume Ratio-----	15
2. Preliminary Valve Layout-----	11
3. Preliminary Cylinder Head Design---	25
4. Offset Cylinder-----	32
5. Single-cam Valve Gear-----	36
6. Bell-crank Valve Gear-----	37
7. Spur Gears Inside Bearing-----	39
8. Cycle with Vacuum Exhaust-----	57-A
9. Precompression Cycle-----	57-B
10. Normal Cycle-----	57-C
11. Exaggerated Vacuum-exhaust Cycle---	57-D
12. Exaggerated Precompression Cycle---	57-D
13. Accepted Valve Gear-----	41

CHAPTER I

1. The first settlement in America was made by the Spaniards in 1492.
2. The English first settled in America in 1607.
3. The French first settled in America in 1608.
4. The Dutch first settled in America in 1614.
5. The Swedish first settled in America in 1638.
6. The Danish first settled in America in 1671.
7. The German first settled in America in 1683.
8. The Irish first settled in America in 1691.
9. The Scottish first settled in America in 1703.
10. The Welsh first settled in America in 1715.
11. The English first settled in America in 1727.
12. The French first settled in America in 1739.
13. The Dutch first settled in America in 1751.
14. The Swedish first settled in America in 1763.
15. The Danish first settled in America in 1775.
16. The German first settled in America in 1787.
17. The Irish first settled in America in 1799.
18. The Scottish first settled in America in 1811.
19. The Welsh first settled in America in 1823.
20. The English first settled in America in 1835.

List of Illustrations, Continued.

Plates.

1. Cylinder Layout
2. Detail of Frame.
3. Valve-gear Layout
4. Detail of Cylinder
5. Details of Valve-gear Mechanism
6. Details of Flywheel, Crankshaft, and
Reciprocating Parts.
7. Miscellaneous Details.

THE HISTORY OF THE	1
REIGN OF	2
THE HISTORY OF THE	3
REIGN OF	4
THE HISTORY OF THE	5
REIGN OF	6
THE HISTORY OF THE	7
REIGN OF	8
THE HISTORY OF THE	9
REIGN OF	10
THE HISTORY OF THE	11
REIGN OF	12
THE HISTORY OF THE	13
REIGN OF	14
THE HISTORY OF THE	15
REIGN OF	16
THE HISTORY OF THE	17
REIGN OF	18
THE HISTORY OF THE	19
REIGN OF	20
THE HISTORY OF THE	21
REIGN OF	22
THE HISTORY OF THE	23
REIGN OF	24
THE HISTORY OF THE	25
REIGN OF	26
THE HISTORY OF THE	27
REIGN OF	28
THE HISTORY OF THE	29
REIGN OF	30
THE HISTORY OF THE	31
REIGN OF	32
THE HISTORY OF THE	33
REIGN OF	34
THE HISTORY OF THE	35
REIGN OF	36
THE HISTORY OF THE	37
REIGN OF	38
THE HISTORY OF THE	39
REIGN OF	40
THE HISTORY OF THE	41
REIGN OF	42
THE HISTORY OF THE	43
REIGN OF	44
THE HISTORY OF THE	45
REIGN OF	46
THE HISTORY OF THE	47
REIGN OF	48
THE HISTORY OF THE	49
REIGN OF	50
THE HISTORY OF THE	51
REIGN OF	52
THE HISTORY OF THE	53
REIGN OF	54
THE HISTORY OF THE	55
REIGN OF	56
THE HISTORY OF THE	57
REIGN OF	58
THE HISTORY OF THE	59
REIGN OF	60
THE HISTORY OF THE	61
REIGN OF	62
THE HISTORY OF THE	63
REIGN OF	64
THE HISTORY OF THE	65
REIGN OF	66
THE HISTORY OF THE	67
REIGN OF	68
THE HISTORY OF THE	69
REIGN OF	70
THE HISTORY OF THE	71
REIGN OF	72
THE HISTORY OF THE	73
REIGN OF	74
THE HISTORY OF THE	75
REIGN OF	76
THE HISTORY OF THE	77
REIGN OF	78
THE HISTORY OF THE	79
REIGN OF	80
THE HISTORY OF THE	81
REIGN OF	82
THE HISTORY OF THE	83
REIGN OF	84
THE HISTORY OF THE	85
REIGN OF	86
THE HISTORY OF THE	87
REIGN OF	88
THE HISTORY OF THE	89
REIGN OF	90
THE HISTORY OF THE	91
REIGN OF	92
THE HISTORY OF THE	93
REIGN OF	94
THE HISTORY OF THE	95
REIGN OF	96
THE HISTORY OF THE	97
REIGN OF	98
THE HISTORY OF THE	99
REIGN OF	100

List of Symbols.

I.H.P. , rated indicated horsepower.

E , mechanical efficiency.

B.H.P. , rated brake horsepower.

f_o , overload factor.

f_r , rating factor.

B.H.P._{max} , maximum brake horsepower.

I.H.P._{max} , maximum indicated horsepower.

V_1 , volume of charge at beginning of compression.

V_2 , volume of charge at end of compression = clearance volume.

r , ratio of volumes at beginning and end of compression.

p_1 , abs. press. at beginning of compression.

p_2 , abs. press. at end of compression.

n , exponent for compression.

v , piston speed.

L , length of stroke.

d , diameter of bore of cylinder.

N , number of explosions per minute.

REIGN OF

CHARLES THE FIRST

BY

JOHN BURNET

OF

THE UNIVERSITY OF OXFORD

IN TWO VOLUMES

THE SECOND VOLUME

OF THE HISTORY OF THE

REIGN OF

CHARLES THE FIRST

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IN TWO VOLUMES

THE SECOND VOLUME

OF THE HISTORY OF THE

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JOHN BURNET

P_{max} , mean effective pressure maximum.

A , area of cylinder bore.

n' , revolutions per minute.

D , piston displacement.

t , thickness of cylinder wall.

p , maximum explosion pressure.

S , allowable stress in cast iron.

C , constant used in cylinder design.

t_j , thickness of jacket wall.

d_i , diameter of inlet port.

d_e , diameter of exhaust port.

w_i , width of inlet-valve seat.

d_1 , root diameter of cylinder-head bolt.

n_1 , number of cylinder-head bolts.

P , maximum thrust against head.

p_t , allowable stress in bolts.

d_E , excess energy absorbed by flywheel.

W , weight of flywheel.

g , acceleration due to gravity.

K , constant in flywheel design.

ρ , radius of gyration of flywheel.

$=$ (approximately) radius of flywheel.

d_c , diameter of crank pin.

l_c , length of crank pin.

L' , length of piston.

p_n , net bearing pressure of piston against wall.

t_1 , thickness of piston barrel.

t_r , thickness of piston ring.

w , width of piston ring.

n_r , number of piston rings.

l_2 , length of piston pin.

d_2 , diameter of piston pin.

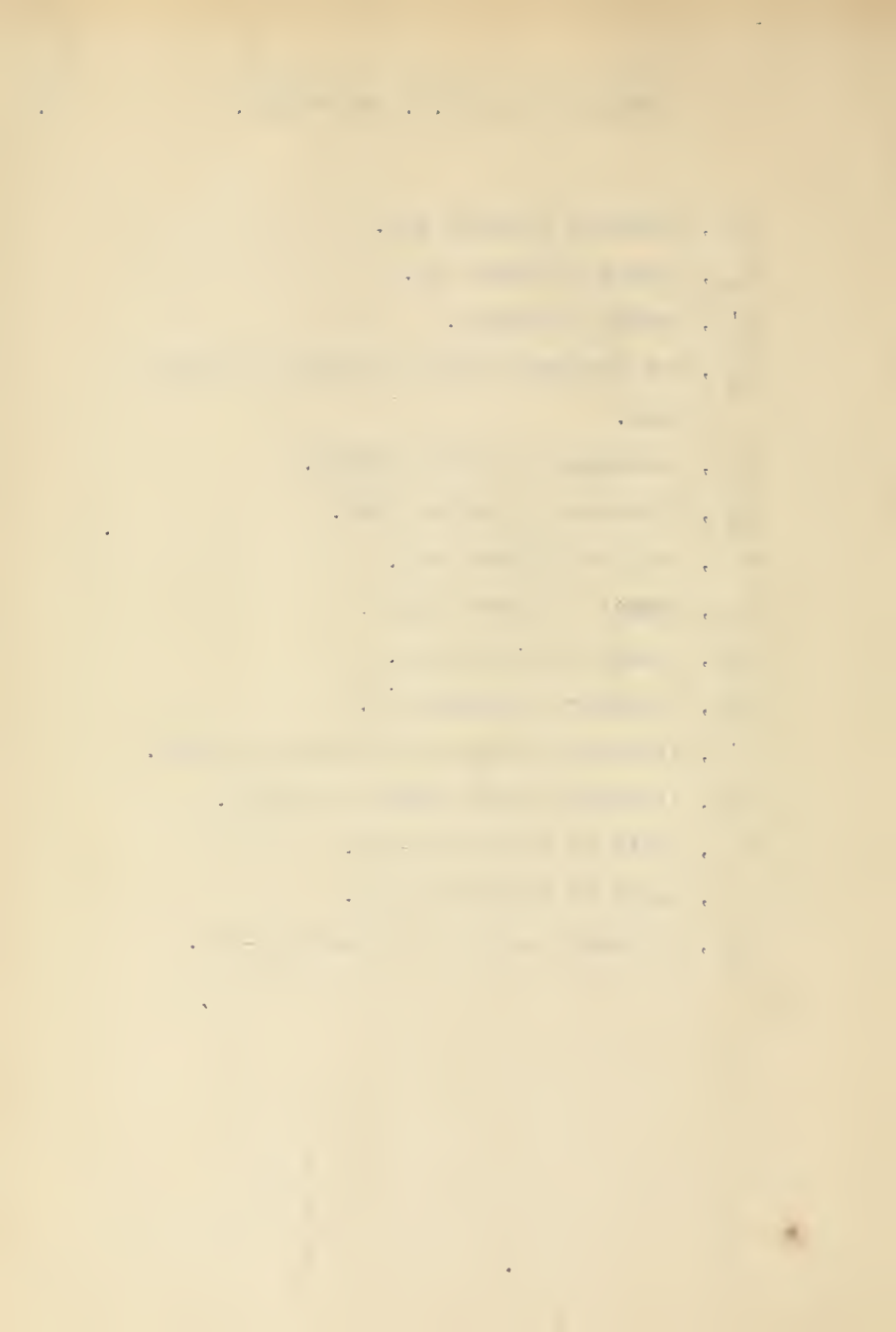
l' , distance between set screws on piston.

d_s , diameter of set screws on piston.

a , width of connecting-rod.

b , depth of connecting-rod.

p_c , allowable stress in connecting-rod.



THE DESIGN-STUDY OF A 10 H.P., HIGH-
DUTY STATIONARY GAS ENGINE.

OBJECT and METHOD.

Object:-

It was desired to design, build, and perfect a small gas engine of the stationary type which could be used in the laboratories of the Armour Institute of Technology for experimental purposes of a varied nature. The prime requisites of this engine are: (a) high-duty, i.e. high power for the particular piston displacement, which will permit of a large overload factor; (b) lightness, to result from the proper proportioning of the parts, especially the base and cylinder; and (c) simplicity in the valve-gear, to provide a ready and easy method of adjusting the valves.

In this thesis it was the object to concentrate attention upon the most important points in the design of the engine, and to leave the building, perfecting, and testing as subjects for further thesis-investigations by students of succeeding classes in mechanical engineering.

Method:-**I. Preliminary Considerations.**

At the outset it was necessary to make assumptions of the most important design constants, and to decide upon the type of engine, fuel to be used, speed, and various other points. These considerations and the various values and assumptions decided upon may be tabulated as follows:

10 h.p. (rating) at 300 r.p.m.

Single, horizontal, overhung cylinder.

Four-stroke cycle.

Three valves- two exhaust at bottom and one inlet at top. (Fig. 2).

Valves set at 45 degrees to the horizontal. (Fig. 2).

Offset cylinder.

Center crank.

Two flywheels.

Fuel-gasoline or illuminating gas.

Flywheel, throttle governor.

Mean effective pressure maximum, 125 lbs.

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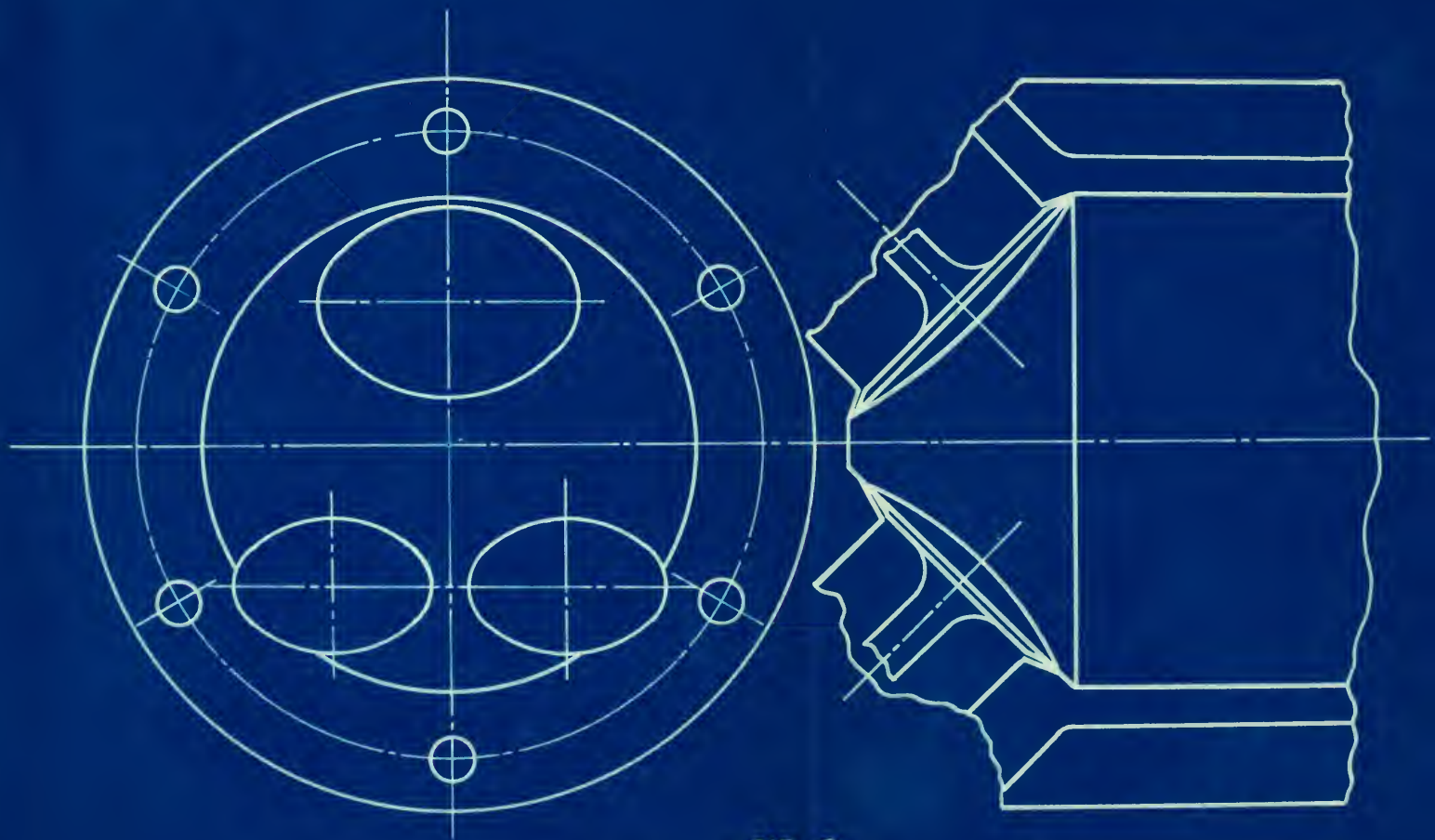


FIG. 2.
PRELIMINARY VALVE LAYOUT.

II. Preliminary Calculations.

Before the actual layout of the engine upon the drawing board could be started, it was necessary to calculate various data and at the same time to make further and more specific assumptions, all of which would finally lead to the determination of the required bore and stroke, and clearance volume.

(a) Rated Indicated Horsepower:

Let I.H.P. = rated indicated horsepower

E = mechanical efficiency

B.H.P. = rated brake horsepower = 10.

Then $I.H.P. = B.H.P. / E$.

Since at the outset a high-grade engine only was considered, the relatively high mechanical efficiency of 90 per cent may be assumed.

$$\text{Therefore, } I.H.P. = \frac{10}{0.90} = 11.1 \text{ h.p.}$$

(b) Maximum Brake and Indicated horsepower.

An overload factor of 50 per cent,

which is equivalent to a rating factor of $66\frac{2}{3}$ per cent, was assumed. This high overload factor was used, because it was believed that this engine, with its high volumetric and thermal efficiencies, could develop a maximum mean effective pressure of 125 lb. per sq.in., whereas the ordinary engine of this type is capable of only 105 lb. per sq.in. maximum mean effective pressure.

Let f_o = overload factor = 50%.

f_r = rating factor = $66\frac{2}{3}$ %.

B.H.P._{max} = maximum brake horsepower

I.H.P._{max} = maximum indicated horsepower

Then B.H.P._{max} = B.H.P. $(1+f_o)$ = $10 \times (1+.50)$ =
15 horsepower.

also B.H.P._{max} = $\frac{\text{B.H.P.}}{f_r} = \frac{10}{2/3} = 15$ horsepower

And I.H.P._{max} = I.H.P. $(1+f_o)$ = $11.1 \times (1+.50)$ =
16.65 horsepower.

also I.H.P._{max} = $\frac{\text{I.H.P.}}{f_r} = \frac{11.1}{2/3} = 16.65$ h.p.

The first settlement in Boston was made in 1630 by a group of Puritan settlers from England. They came to the city in search of religious freedom and a place to practice their faith. The settlers were led by John Winthrop, who gave them the name "Boston" in honor of the city of Boston in England. The city grew rapidly and became one of the most important centers of commerce and industry in the New England region. In 1780, the city was the site of the Battle of the Clouds, a significant battle in the American Revolutionary War. The city was then the capital of the Commonwealth of Massachusetts. In 1830, the city was the site of the first public school in the United States. The city has since become one of the most important centers of commerce and industry in the United States. It is home to many of the most famous universities and research institutions in the world. The city is also known for its rich cultural heritage and its many historic landmarks. Today, the city is a vibrant and diverse community, and it continues to be one of the most important centers of commerce and industry in the United States.

(c) Ratio of Volumes at Beginning and End of Compression (Fig. 1):

The vacuum during the suction stroke was taken as 1.5 lb. for good filling, and the compression pressure assumed for illuminating gas or gasoline was 85 lb. gauge. The value of "n" to be used in the computation for compression was 1.31 (for ordinarily good cooling).

Let r = ratio of volumes at beginning and end of compression.

V_1 = volume of charge at beginning

V_2 = volume of charge at end (i.e. clearance volume)

p_1 = abs. press. at beginning = 13.2 lb.

p_2 = abs. press. at end = 99.7 lb.

n = exponent for compression = 1.31

$$\text{Then } r = \frac{V_1}{V_2} = \frac{p_2}{p_1}^{\frac{1}{n}} \quad (\text{from the fundamen-})$$

tal relation- $p_1 V_1^n = p_2 V_2^n$ for a perfect gas)



$$\text{and } r = \frac{99.7}{13.2} \frac{1}{1.31} = (7.55)^{.764} = 4.675$$

(d) Percent of Clearance:

(1) Based upon piston displacement.

$$= \frac{V_2}{V_1 - V_2} = \frac{1}{\frac{V_1}{V_2} - 1} = \frac{1}{r-1} = \frac{1}{4.675-1} = \frac{1}{3.675}$$

$$= 27.21 \text{ per cent.}$$

(2) Based upon piston displacement

$$\text{clearance} = \frac{27.21}{127.21} = 21.4 \text{ per cent.}$$

(e) Bore and Stroke:

The following assumptions and values previously found by calculation were employed in the calculation of the required bore and stroke:

$$E = \text{mechanical efficiency} = 90\%.$$

$$\text{B.H.P.}_{\text{max}} = 15 \text{ h.p.}$$

$$v = \text{piston speed in ft. per min.} = 500$$

$$\frac{L}{d} = \text{ratio length of stroke to bore} = 1.5$$

The new symbols used are:

L = length of stroke in inches.

d = diameter of bore of cylinder in inches.

N = number of explosions per minute.

$P_{\max}$ = mean effective pressure, maximum
(= 125 lb.)

A = area of bore in sq.in.

n' = revolutions per minute.

From the relation between piston speed and revolutions of the crank shaft.

$$v = 2Ln'$$

$$\text{and } n' = \frac{v}{2L} = \frac{v}{2 \times 1.5d} = \frac{v}{3d}$$

For the four-cycle, single-cylinder engine

$$N = \frac{n'}{2} = \frac{v}{2 \times 3d} = \frac{v}{6d}$$

The maximum brake horsepower is:

$$\begin{aligned} \text{B.H.P.}_{\max} &= \text{I.H.P.}_{\max} \times E = \frac{P_{\max} \text{ LANE}}{33000} \\ &= \frac{P_{\max} \times 1.5d \times 0.7854d^2 \times \frac{v}{6d} \times E}{33000} \end{aligned}$$

Solving for the diameter of bore:

$$d = \sqrt{\frac{B \cdot H.P._{max} \times 6 \times 33000}{1.5 \times .7854 \times v_{EP_{max}}}}$$

$$= \sqrt{\frac{15 \times 6 \times 33000}{1.5 \times 0.7854 \times 500 \times 0.90 \times 125}} = \sqrt{44.8} = 6.69$$

inches, call it $6\frac{3}{4}$ inches.

The length of stroke is:

$$L = 1.5d = 1.5 \times 6.75 = 10.12 \text{ inches}$$

call it 10 inches

(f) Clearance volume.

Let D = piston displacement in cu.in.

Then the clearance volume is:

$$V_2 = 0.2721 D = 0.2721 \times 0.7854 d^2 L$$

$$= 0.2721 \times 0.7854 \times (6.75)^2 \times 10$$

$$= 0.2721 \times 358.0 = 97.3 \text{ cu.in.}$$

(The piston displacement being found to be 358 cu.in.)

(g) Revolutions per minute

$$n' = \frac{v}{2 \frac{L}{12}} = \frac{500 \times 12}{2 \times 10} = 300 \text{ r.p.m.}$$

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III. Cylinder Design.

A horizontal-type, valve-in-hand engine with overhead camshaft and a detachable cylinder head (Fig. 2.) was decided upon. A general layout drawing of the cylinder, cylinder head, valves and piston was then started. (Plate 1.)

A $3\frac{1}{2}$ inch diameter inlet port and two- $2\frac{1}{2}$ inch diameter exhaust ports were used. (Fig. 2.) These dimensions will insure good filling and scavenging, since at a piston speed of 500 ft. per minute the velocity of the gases will be under 50 ft. per second. If only one of the exhaust valves is used at a particular part of a stroke of the piston (see p.55.), the velocity of the burnt gases will not exceed 65 ft. per second. It is evident, therefore, that the valve-ports are much larger than is ordinarily the case with this type of motor. Valve-seats of 30 degrees were chosen, in accordance with modern practice, to give a larger area of opening;

ORIGINAL ARTICLES

- 1. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1111)
- 2. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1112)
- 3. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1113)
- 4. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1114)
- 5. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1115)

(Continued on page 1116)

ORIGINAL ARTICLES (Continued)

- 6. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1116)
- 7. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1117)
- 8. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1118)
- 9. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1119)
- 10. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1120)

The following articles are also included in this issue:

11. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1121)

12. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1122)

13. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1123)

14. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1124)

15. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1125)

16. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1126)

17. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1127)

18. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1128)

19. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1129)

20. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1130)

21. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1131)

22. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1132)

23. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1133)

24. THE EFFECT OF VARIOUS DOSES OF VITAMIN B₁ ON THE GROWTH OF RATS. (P. 1134)

for small valve-lifts.

The main points of the cylinder and head containing the valves were then sketched in; the following calculations being necessary in order to determine important dimensions:

a) Thickness of cylinder wall:

Let t = thickness of wall in inches

p = maximum explosion pressure = 450 lb.
per sq.in.

d = diameter of bore in inches = 6.75 in.

S = allowable stress in cast iron = 3500
lb. per sq.in.

and C = constant to allow for cooling and other stresses = 0.3 inches.

$$\begin{aligned}\text{Then } t &= \frac{pd}{2S} + C = \frac{450 \times 6.75}{2 \times 3500} + 0.3 \\ &= 0.434 + 0.3 = 0.734\end{aligned}$$

say $\frac{3}{4}$ inch diameter.

This formula for the thickness consists of two parts- the theoretical formula plus a practical constant, "C". Haeder and

Huskisson, and Gldner in their treatises on gas engine design give the following formula, which results in a slightly larger value:

$$t = \frac{d}{2} \left(\sqrt{\frac{S+0.4p}{S-1.3p}} - 1 \right) + C$$

where $C = 0.375$ for engines of this size.

$$\begin{aligned} t &= \frac{6.75}{2} \left(\sqrt{\frac{3500 + 0.4 \times 450}{3500 - 1.3 \times 450}} - 1 \right) + 0.375 \\ &= \frac{6.75}{2} \left(\sqrt{\frac{3680}{2915}} - 1 \right) + 0.375 \\ &= 0.412 + 0.375 = 0.787 \end{aligned}$$

say $\frac{3}{4}$ inch diameter.

b) Thickness of jacket wall:

Haeder and Huskisson give $\frac{9}{16}$ inch as a practical dimension for the thickness of the jacket wall for 9-11 h.p. engines.

The instruction sheets on gas engine design used at Armour Institute of Technology give an empirical formula for the value of t_j , the thickness of the jacket wall:

$$t_j = 0.06d \text{ to } 0.10d$$

using $t_j = 0.08d$

$$= 0.08 \times 6.75 = 0.54$$

$$\text{say } \frac{9}{16} \text{ inch.}$$

c) Water space:

$$\text{A practical value of } \frac{15}{16} \text{ inch as given}$$

by several authorities was used.

d) Diameters of valve-stems:

Let d_i = diameter of inlet port

and d_e = diameter of exhaust port

$$\text{Diam. of inlet-valve stem} = 0.14d_i = 0.14 \times 3.5$$

$$= 0.49 \text{ inch}$$

$$\text{Diam. of exhaust-valve stems} = 0.14d_e = 0.14$$

$$\times 2.5 = 0.35 \text{ inch}$$

$$\text{say } \frac{1}{2} \text{ and } \frac{3}{8} \text{ inch respectively}$$

(According to Levin)

e) Width of valve seat

The width of the inlet-valve seat is

$$= 0.01d_i + 3/16$$

$$= 0.01 \times 3.5 + 0.1875 = 0.222 \text{ inch}$$

$$\text{say } \frac{1}{4} \text{ inch (According to Streeter)}$$

The actual width was made $3/8$ inch on all three valve-seats.

f) Cylinder-head bolts:

Let d_1 = root diameter of bolt

n_1 = number of bolts (assumed as six)

p = maximum thrust against head

p_t = allowable stress in bolts

The maximum thrust against the head is:

$$\begin{aligned} p &= \pi/4 d^2 p_t \\ &= 0.7854 \times (6.75)^2 \times 450 = 16,100 \text{ lb.} \end{aligned}$$

The resisting stress in the bolts is:

$$\begin{aligned} \pi/4 d_1^2 p_t n_1 &= P \\ \text{and } d_1 &= \sqrt{\frac{4p}{\pi p_t n_1}} \end{aligned}$$

where p_t can be taken as 13,500 lbs. per sq. in. by using 3.5% nickel steel (0.40% carbon) bolts.

$$\text{Then } d_1 = \sqrt{\frac{4 \times 16,100}{3.1416 \times 13,500 \times 6}} = \sqrt{0.253}$$

= 0.504 in. root diam. of bolt.

The nominal diameter is $5/8$ inch.

Because of the complicated shape of the cylinder head, its theoretical design was of little or no importance; and was at best unsatisfactory. The design, therefore, was carried out from an empirical standpoint. The thickness of the inside wall or dome was made the same as that of the cylinder wall.

The exhaust ports were located in the bottom part of the cylinder head and were separated by the water jacket to secure proper cooling of the exhaust valves. These two ports lead to an exhaust manifold at the bottom of the cylinder head. The inlet port was at first designed to lead in from the top of the cylinder head (Fig. 3.); but in order to avoid trouble in the location of the carburetor, this was changed to lead in at the side of the cylinder head.

In the attempt to secure as nearly as possible a hemispherical combustion chamber, considerable difficulty was met because it



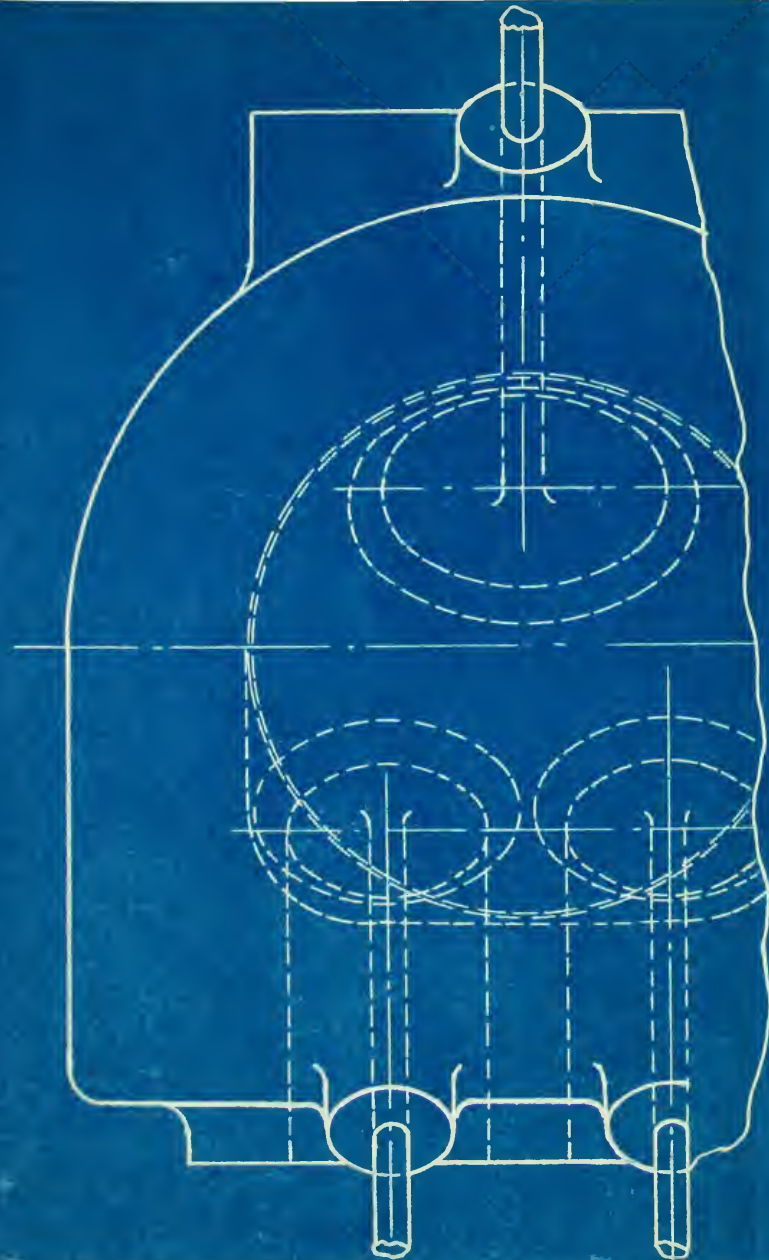


FIG. 3.
PRELIMINARY CYLINDER HEAD DESIGN.

was necessary to have the exhaust-valve stems (and consequently the valve faces) in the same plane. The shape of the combustion chamber was shown in the drawings as clearly as possible; but in order to have this correct in the engine, great care and supervision will have to be exercised in the making of the pattern.

The cylinder head was made flat on the end (Fig. 3.), in order to give sufficient water-space for thoroughly cooling the head and valves and in order to make it more convenient in bolting the head to the cylinder and the valve-gear brackets to the cylinder head.

Other points given especial consideration in the design of the cylinder were:- the locations of cylinder-head bolts, spark-plug tap, inlet and outlet jacket-water taps, and indicator tap, the design of valve-stem guides, the bolting of the cylinder on the frame from the inside of the crankcase, and the bolting of the cylinder head and cam-shaft bearing brackets to the cylinder.

IV. Frame Design.

In the design of the frame, it was considered more economical and also more practical to construct a frame and base in one piece.

The frame was designed empirically, sufficient thickness of material being provided for at the proper points. As the cylinder was to be overhung, the front wall of the frame was made 1 inch thick in order to allow for a counterbore of about $\frac{3}{16}$ inch to line-up the cylinder.

The length of the frame from the center line of the main bearing to the edge of the supporting wall for the cylinder, was determined by the length of the connecting-rod, the length of the stroke, the position of the piston pin, and also the extreme positions of the piston.

As the main-bearing was also designed empirically, it was proportioned to conform properly to the crank-shaft, which is $2\frac{3}{4}$ inches in diameter. The length of the journal was made $6\frac{1}{4}$ inches, including bosses on the ends. The

use of ball-bearings was considered. These may be probably added later in analyzing the stability, smoother running and decrease of vibration of the engine.

The bearing cap was made in accordance with common practice. The cap is to be held on by four bolts. A sight feed oil cup will be set in the center of the cap. The bearing has a suitable oil well to take care of the oiling of the crank shaft.

Proper pads or bosses were designed to take care of the bearings of the valve mechanism, which consists of a horizontal lay shaft running along the side of the frame and cylinder to the cam shaft which is perpendicular to the axis of the cylinder.

In designing the base, the diameter of the flywheel was taken into consideration. It was thought that to make the base a trifle deeper would be cheaper than to dig pits in the floor so that the flywheels could run clear.

The sides of the base were made at a slight angle to the vertical. The base was ribbed to secure solid support and stability..Six 1-inch bolts were found sufficient to hold the engine down. Pads, supported by webs, were added at the points where the bolts were placed.

It will be noticed by examining the drawings of the frame and base that the size has been reduced to a minimum. The object was to obtain a long narrow construction; and a fair degree of success was met in that direction. At no point is the metal thicker than 1-inch or thinner than $\frac{1}{2}$ inch. The frame in general is compact, small, and of sufficient strength to take care of all the stresses to which it will be subjected at maximum load.

One of the first considerations in the design of the engine was the offsetting of the cylinder. After making several sketches and studying the appearance of the construction, it was decided to make the center line

of the crankshaft 1 inch, or one tenth the length of the stroke, below the center line of the cylinder.

The object of offsetting the cylinder may be briefly stated as follows:-

"The side thrust of the piston against the cylinder wall is the cause of both wear of the piston and cylinder and of frictional loss, and any means of reducing it is therefore doubly advantageous. The most obvious way of accomplishing the reduction of the effective pressure on the piston is by making the connecting-rod longer. In practice, however, the length of the connecting-rod is limited, since the bulk and weight of the engine increase rapidly with this length. There is, however, a method of arranging the moving parts so that with a fixed length of connecting-rod the connecting-rod angle will be relatively small when the pressure on the piston is high, and proportionably greater when the pressure

on the piston is low. This consists in offsetting the crankshaft axis from the axis through the cylinder toward the side on which the crank arm moves during the power stroke. The angularity of the connecting-rod for any position on the power stroke, except at the very ends of the stroke, is then less than it is for the corresponding position in an engine without an offset cylinder.

Offsetting the cylinder has some rather peculiar effects on the functions of the engine. In the first place, the piston is not at the end of its stroke when the crank arm is in the vertical positions; it reaches the end of the stroke a moment later, when the crank arm and connecting-rod are in line with each other. Besides, the two dead center positions are not 180 degrees apart, as they are in a symmetrical engine, but slightly less, as shown in the Fig. 4.

The effective or actual length of the



FIG. 4.
OFFSET CYLINDER.

stroke, instead of being equal to twice the length of the crank arm, is slightly greater than this. The connecting-rod is parallel with the cylinder axis some time after the inner crank center position has been passed, and the same time before the head center position is reached.

The length of the offset employed in American practice is equal to one-sixth of the bore."

Heldt, P.M., "The Gasoline Automobile",

Vol.I.

Main Bearings and Bearing Cap.

The main bearings of an engine are as important as the shaft, and should be constructed with the same care given the shaft. There is nothing about the bearings that can be designed rationally after the diameter and length have been computed.

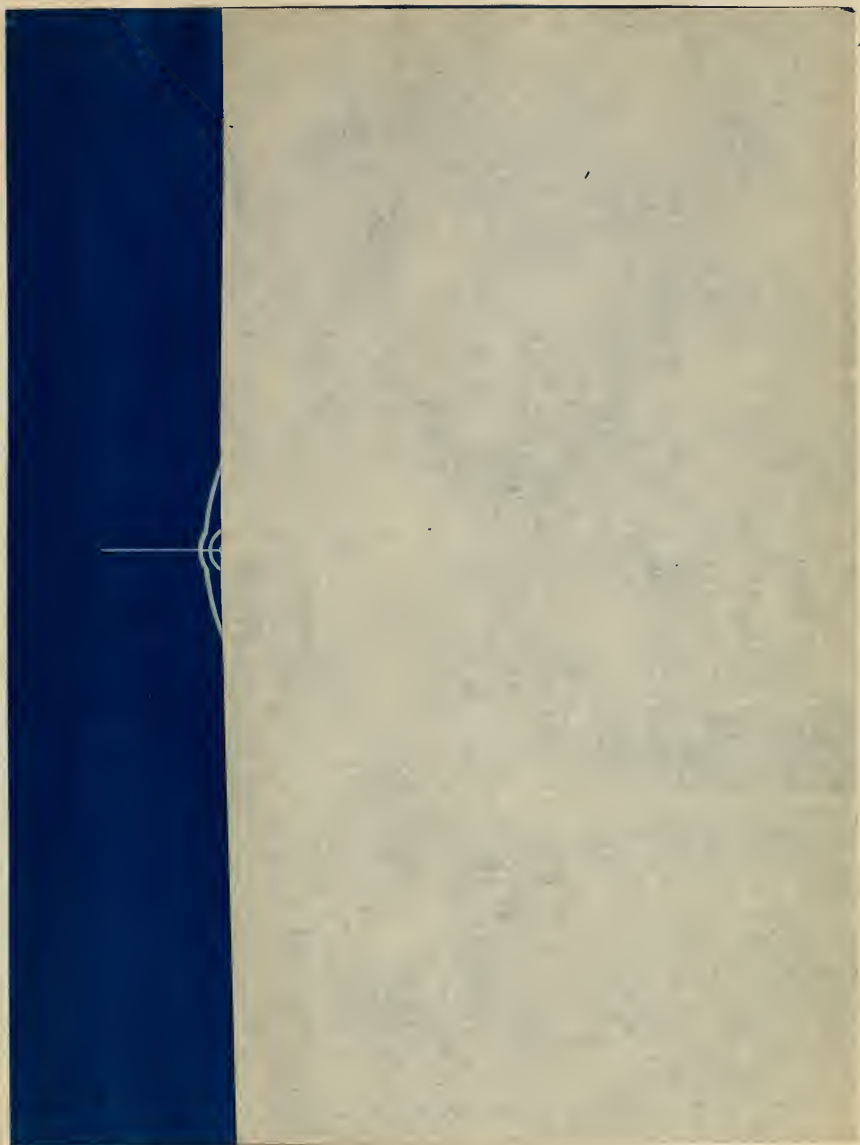
A bearing with an horizontal cap was selected as shown in Plate 7. The bearing cap is bolted to the frame of the engine by means of four bolts.

Future investigators on this engine may well consider the use of ball-bearings to replace either the babbit or bronze bushings.

V. Valve Gear.

Before the final adoption of the valve-gear arrangement most suitable for this type of engine, several other forms were considered. The single-cam gear shown in the sketch (Fig. 5.) was at first considered. The valves are opened by a rocking lever pivoted at the center to a bracket fastened to the cylinder head. The lower end of this rocker is connected to a push rod which is operated by the cam, whose shaft is turned by a pair of 2-1 spur gears from the crank shaft. The push rod and cam are under the cylinder.

Another type of gear to be studied was that shown in the sketch (Fig. 6) A motion of oscillation is transmitted to the bell crank by means of an eccentric and long connecting valve rod, the eccentric shaft being geared to the crank shaft through a pair of 2-1 spur gears. A roller on the pin through



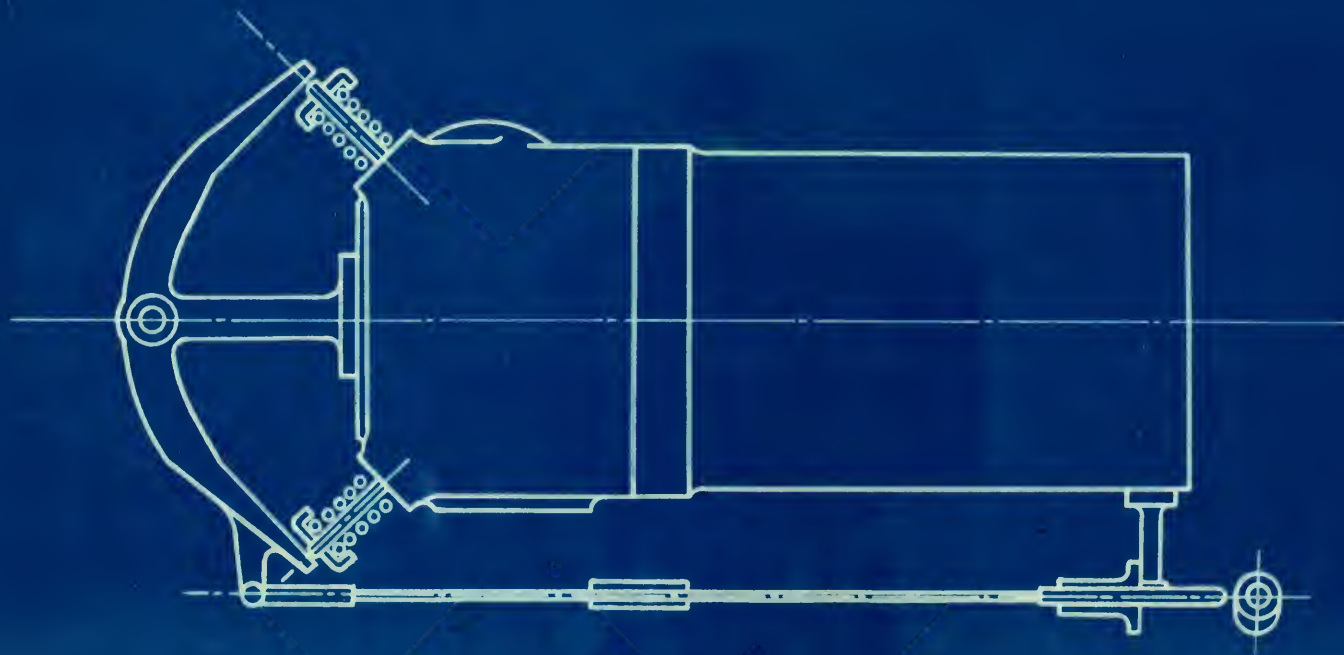


FIG. 5.
SINGLE-CAM VALVE GEAR.



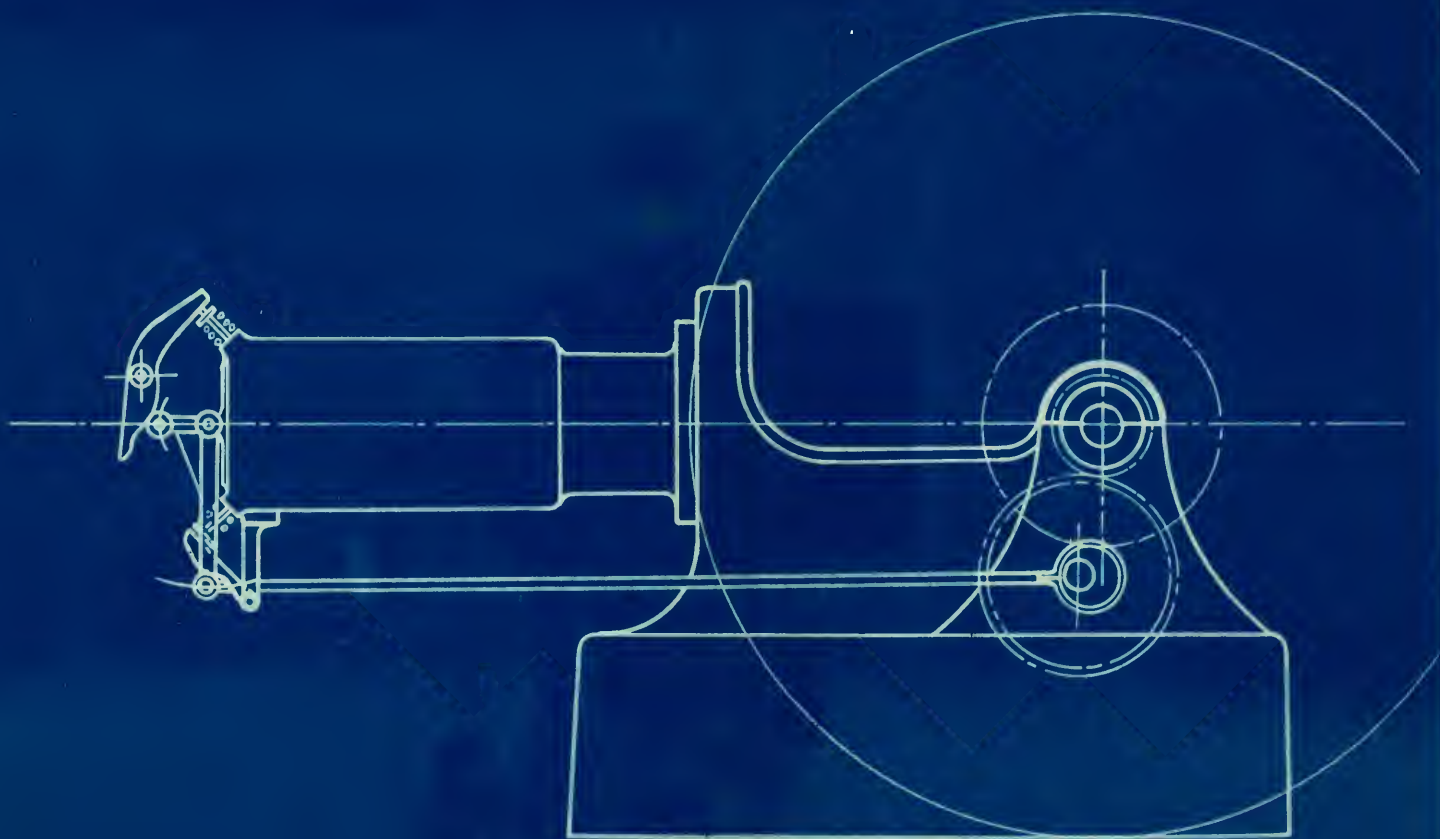


FIG. 6.
BELL-CRANK VALVE GEAR.

the lower arm of the bell crank operates the exhaust-valve, rocker arm, and a similar roller on the upper arm of the bell crank operates the inlet-valve, rocker arm.

As was originally planned, the overhead cam-shaft type of valve gear was finally decided upon and designed to meet the peculiar requirements of this engine. The spur gears driving the jack shaft were made 5 inches and 10 inches in pitch diameter, respectively, with a diametral pitch of 6 and width of face of $1\frac{1}{8}$ inches. They were at first placed inside of the bearing as shown in sketch, Fig. 7 ; but because of their inaccessibility and the trouble in providing a proper and sufficient bearing for the jack shaft, it was decided to place these gears between the bearing of the crank shaft and the flywheel. The jack shaft was placed at a distance below the center-line of the cylinder such that the horizontal centerline of the lay shaft coincided

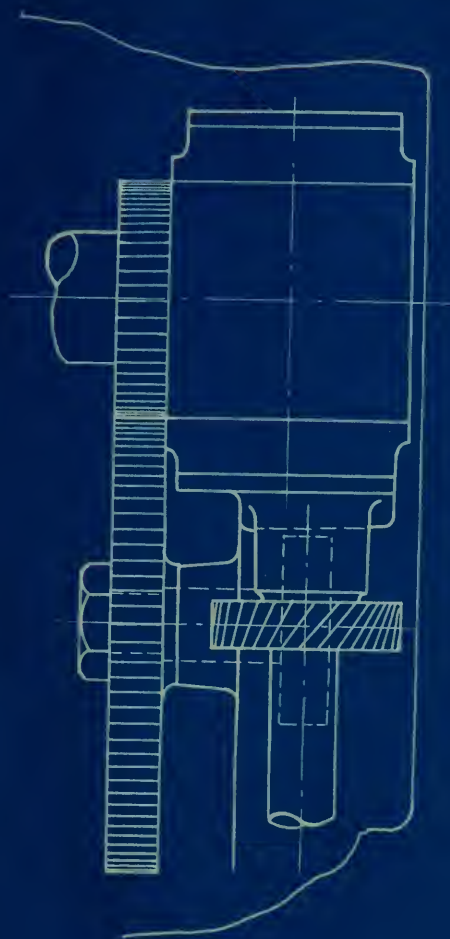


FIG. 7.
SPUR GEARS INSIDE BEARINGS

with the horizontal center-line of the cylinder. The spiral gears were made 4 inches pitch diameter with a 1- inch face. The bevel gears were made 4 inches pitch diameter, 8 diametral pitch, with a 1-inch face.

In designing the rocker arms and brackets, considerable difficulty was encountered in securing the proper location of parts. Proper length of valve stem for the springs was first secured; and then the rocker arms were drawn in at right angles to the valve stems. The lift of the valves, the size of rollers on the rocker arms, the diameter of the cam shaft and the size of the cams were all determining factors in the location of the cam shaft.

A single-piece bracket was used for the cam-shaft and rocker-arm bearings. Although this casting is complicated and requires considerable machine work, it was thought advisable to make it in one piece to secure rigidity and to maintain alignment and proper valve-timing.



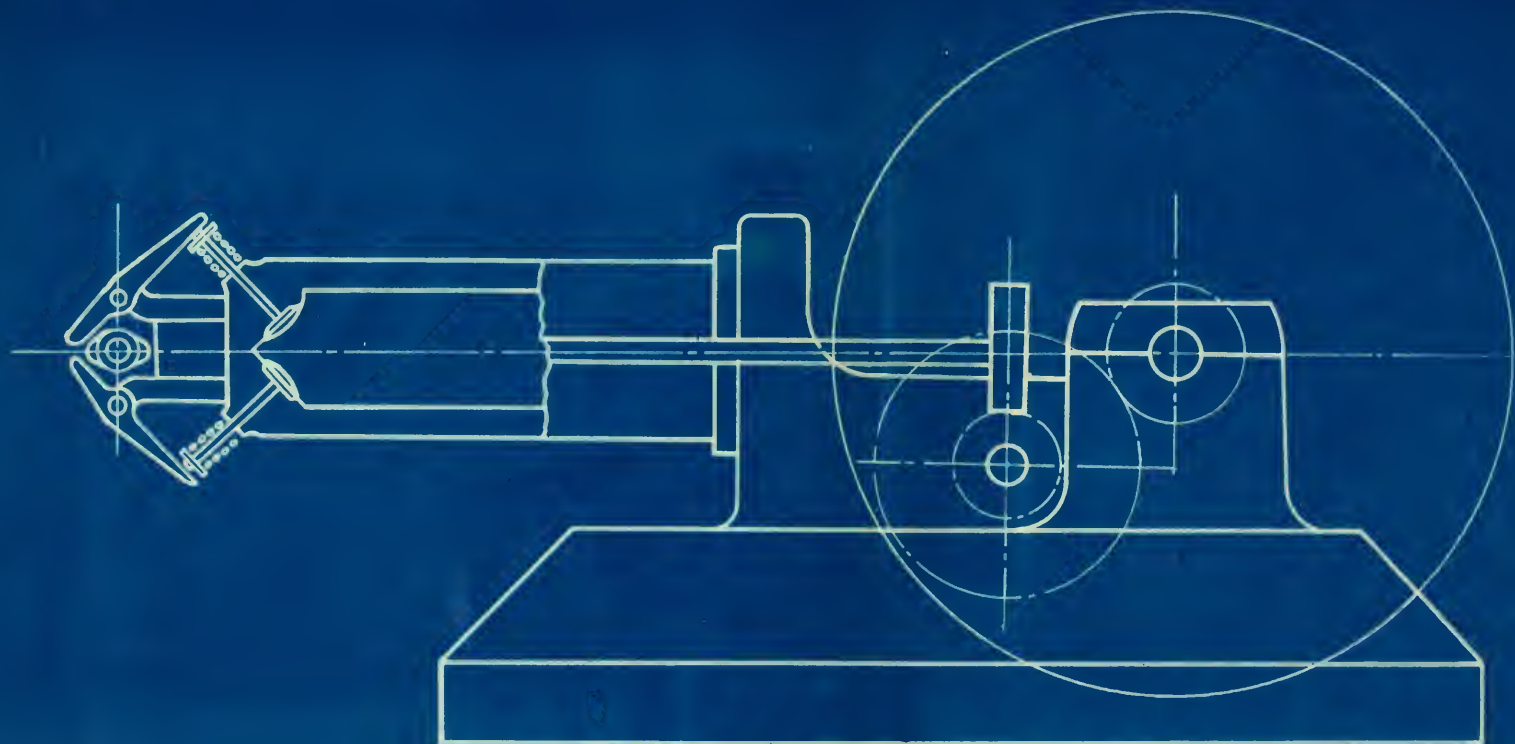


FIG. 13.
ACCEPTED VALVE GEAR.

The valve springs were designed for the inlet-valve, and the same springs were used for the exhaust valves. A complete vacuum was assumed in the cylinder, and the springs calculated to hold the inlet valve on the seat. Then the total atmospheric pressure against the valve was found to be 144.3 lb. Two concentric springs were decided upon, and springs with the following specifications were found to be correct:

	<u>Outside Spring</u>	<u>Inside Spring</u>
Outside diam.	1½ inch	1 inch
Wire	No.6,0.192 in.	No.10,0.135 in.
Allowable load	128 lb.	67 lb.
Deflection per coil		
per 100 lb. load	0.108 inch	0.135 inch

It was assumed that the outside spring would carry the entire load of 144 lb.

Deflection per coil per 100 lb. load

= 0.108 inch.

Deflection per coil per 144 lb. load is

$$\frac{144}{100} \times 0.108 = 0.156 \text{ inch}$$

$$\text{The number of coils is } \frac{0.875}{0.156} = 6 \text{ coils}$$

($7/8$ inch = 0.875 inch = lift of inlet valve.)

For the sake of safety, 8 coils were used on the outside spring and 11 coils on the inside spring. The length of the springs when open was made $2-5/8$ inches. The method of securing the spring is clearly shown in Plate III.

VI. Flywheel.

The flywheel should be designed by an analysis of the inertia forces, rotative efforts, and the like. This was too extensive to be included in this work, so empirical formulae and constants used in practice were employed in the design of the flywheel.

The approximate weight of the flywheel was found as follows:

The excess energy, dE , which must be absorbed and given out by the flywheel bears some relation to the work generated per revolution or per cycle in the engine cylinder.

This ratio $\frac{\text{Energy absorbed by flywheel}}{\text{Indicated work per revolution}}$

$$= \frac{dE}{\frac{\text{I.H.P. MAX}}{n'}} \quad \text{is practically constant for a}$$

given type of engine.

A good value for this ratio for a single-

acting four-cycle engine is 2.4 (Streeter).

Indicated work per revolution is

$$\frac{\text{I.H.P.}_{\text{max}} \times 33000}{n'} = \frac{16.65 \times 33000}{300}$$

$$\text{Then } dE = \frac{2.4 \times 33000 \times 16.65}{300} = 4400 \text{ ft.lb.}$$

A coefficient of fluctuation of 1 percent was assumed. Then the weight of the flywheel

$$W = \frac{dE \times g \times 900 \times k}{\pi^2 \rho^2 n'^2} = \frac{4400 \times 32.2 \times 900 \times 100}{\pi^2 \times 2 \times 2 \times 300 \times 300} = 359 \text{ lb.}$$

ρ = the radius of gyration of the flywheel. The diameter of the flywheel was assumed as 4 feet. With this weight, a flywheel rim was drawn which looked substantially well-proportioned and of sufficient strength. The arms were made elliptical in cross-section and tapering from the hub to the rim.

The hub was to be split, and four bolts were provided for to hold the flywheel on the shaft. The flywheel was also to be secured by a rectangular key. The bore was to be grooved

so that machining a long length of bore would be avoided.

Two flywheels were to be used in order to insure smoother and steadier operation.

Crankshaft.

The engine was to be a center-crank engine with two flywheels, one on each side of the crank-pin. The shaft was designed empirically.

Let d_c = diameter of the pin.

l_c = length of the pin.

d = diameter of cylinder.

Then $d_c = 0.4d = 0.4 \times 6.75 = 2.70 = 2\frac{3}{4}$ inches.

$l_c = 1.4d_c = 1.4 \times 2.70 = 3.78 = 3\frac{7}{8}$ inches.

The journals of the crankshaft were increased slightly in diameter, and sufficient length of shaft was allowed for attaching flywheels, valve gears and brake pulleys.

The crankshaft was to be supported by two bearings and a suitable oiling system was to be supplied.

VII. Reciprocating Parts.

Piston.

In accord with common practice, a trunk-piston was selected. In a single-acting engine this type of piston is usually used, and more often than not it is used without a crosshead.

In order to allow for rings in the piston the total bearing area should be considered to be the product of 0.8 of its length by 0.85 of its diameter. The bearing area is then $0.68dL'$ where d is the diameter and L' is the length of the piston. The bearing pressure should not exceed 30 lb. per sq.in. of this net area. In no case should the length of the piston be less than the diameter. In stationary engines the usual range of values L'/d is $1\frac{1}{4}$ to $1\frac{1}{2}$. A value of $1\frac{1}{3}$ or 1.33 was selected.

Then $L' = 1.33d = 1.33 \times 6.75 = 9$ inches.

Levin gives the following formula for obtaining the net bearing pressure:

Let P = total pressure on piston, in lb.

p_n = net bearing pressure, lb. per sq. in.

$$\begin{aligned}\text{Then } p_n &= \frac{0.1P}{dL'} = \frac{0.1 \times \pi \times 6.75 \times 6.75 \times 450}{4 \times 6.75 \times 9} \\ &= 26.5 \text{ lb. per sq. in.}\end{aligned}$$

This pressure is allowable.

The face of the piston was made concave so that an approximate spherical combustion chamber would be obtained. The thickness of face was made $5/8$ inch. To make the piston stronger the head was ribbed inside with four ribs of a thickness equal to one-half that of the face or $5/16$ inch. The maximum thickness of the piston barrel may be found by the formula.

$$t_1 = (d/50 + t_r + 0.2) \text{ inches.}$$

where d = diameter of cylinder in inches

t_r = thickness of ring in inches

t_1 = thickness of barrel.

$$\text{Then } t_1 = 6.75/50 + 0.25 + 0.2 = 0.585 \text{ inches.}$$

The maximum thickness of barrel was made $5/8$ inch, and the thickness at the end was tapered down to $\frac{1}{4}$ inch.

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The first ring-groove was made at a distance from the end of the piston equal to the thickness of the piston face, that is, $5/8$ in. The thickness of the bridge between the grooves was made equal to the thickness of ring plus $\frac{1}{8}$ inch, or $\frac{1}{2}$ inch. The center line of the piston pin was located 4 inches from the piston face.

Piston Rings.

Plain, cast-iron snap rings, fitted in grooves cut around the circumference of the piston, were made. Three rings were placed ahead of the piston pin, and the fourth ring was so placed that it could serve as a wiping oil ring when the piston reached its outer extreme position. These rings were made of uniform thickness. A pot pattern (see Plate VII) was designed so that two sets of rings could be cut from it. The outside and inside diameters of this finished pattern corresponded to the same dimensions of the piston rings.

Eccentric rings were not used, as they do not present any great advantage. On the other hand, they cost more to make and toward the joint the clearance at the bottom of the groove is increased to a dangerous degree.

From empirical formulae, the dimensions of the rings were calculated.

$$t_r = \text{thickness of piston ring} = \frac{d}{30}$$

$$w = \text{width of piston ring} = \frac{d}{20}$$

Then $t_r = 6.75/30 = .225$ or $\frac{1}{4}$ inch.

and $w = 6.75/20 = .338$ or $\frac{3}{8}$ inch.

The depth of the ring-groove should exceed the thickness of the ring by not more than 0.02 inch. If a greater clearance is allowed, it fills up with oil and the oil thickens, sticking the ring fast in the groove.

$$\text{The number of rings may be found by } n_r = \frac{d}{5w}$$

w = width of ring in inches

d = diameter of piston in inches.

$$\text{Hence } n_r = \frac{6.75}{5 \times .375} = \frac{6.75}{1.875} = 3.6 \text{ or } 4 \text{ rings.}$$

Piston Pin.

The piston pin was designed to carry the maximum total load on the piston, that is, the load due to the explosion pressure. The pin was proportioned so the ratio was $l_2/d_2 = \frac{3}{2}$.

l_2 = length of the piston pin.

d_2 = diameter of the piston pin.

A safe stress of 7000 lb. per sq.in. was assumed, and with a maximum pressure of 450 lb. per sq.in., the diameter of the pin was found to be $1\frac{3}{4}$ inches. The length of the pin was made $3/2$ by $1\frac{3}{4} = 2.625$, or $2\frac{3}{4}$ inches.

The piston pin was made cylindrical in shape, and was to be held in place by two set screws, the positions of which were determined from the relation

$$l' = l_2 + \frac{d - l_2}{2} = \frac{l_2 + d}{2}$$

l' = distance between set screws.

l_2 = length of piston pin.

d = diameter of piston.

$$\text{Then } l' = \frac{2.75 + 6.75}{2} = 4.75 \text{ or } 4\frac{3}{4} \text{ inches.}$$

The diameter of the set screws was determined from the empirical relation

$$d_s = 0.2 d_2 + 3/16 \text{ inch.}$$

d_2 = diameter of piston pin.

d_s = diameter of set screw.

$$\text{Then } d_s = 0.2 \times 1.75 + 3/16 = 0.5375 \text{ or } \frac{1}{2} \text{ inch.}$$

These set screws were to be held in place with lock nuts. This made an inexpensive and convenient method of holding the piston pin and is common practice in small engines with a solid-end connecting-rod.

Connecting-Rod.

Connecting-rods are made of steel forgings or castings. The wrist pin and crank-pin ends should both be made adjustable so as to permit taking up the wear in those bearings.

The connecting-rod of this engine was to be made in three sections. The first section was to be the stub connecting-rod which comprised the small-end (piston end) and the rod, the length of which was almost equal to the entire length of the rod. The second and third sections were parts of the large end (crank-pin end).

The connecting-rod was made equal to 5 crank lengths or 25 inches. The small end was made with a solid eye, containing a bronze bushing and an adjustable wedge. The large head was designed with four bolts, a bronze bushing (split) and copper shim.

The rod proper was calculated merely for alternate tension and compression. The more exact and scientific calculation of the connecting-rod design may be taken up in an advanced thesis. In all probability a connecting-rod will be obtained on the market, at a low price, which will comply with the spec-

ifications of this engine..

The rod was to be of I-beam cross-section. It was designed, however, for a rectangular section.

b = depth of cross-section of rod.

a = width of cross-section of rod.

Assume $b = 1.75 a$.

$$\text{Then } P/p_c = ab = 1.75 a^2. \quad a = \frac{P}{1.75 p_c}$$

P = total load on connecting-rod.

p_c = allowable stress = 6000 lb.per sq.in.

$$\text{Then } a = \sqrt{\frac{16100}{1.75 \times 6000}} = 1.24 \text{ or } 1\frac{1}{4} \text{ inches.}$$

$$\text{and } b = 1.75 \times 1.24 = 2.16 \text{ or } 2\frac{1}{4} \text{ inches.}$$

These dimensions were increased slightly to allow for an I-beam cross-section. The rod was made tapering from the small end to the large end, in depth of rod only, the width remaining constant throughout the length of the rod.

VIII. Means Provided for Experiment.

Besides the ease with which the cams may be adjusted to give any desired valve-setting, several other means of varying the operation of this engine have been provided or proposed.

Although the method of governing has not been considered in this work, this means of controlling and varying the operation is of prime importance. Constant-quality and constant-quantity regulation can be tried, together with means of advancing and retarding the spark. To this end the Pierce Governor as well as flyball and other ring types of governors should be investigated. Under this head the different methods of carburetion should be considered.

Variation in compression will to some extent be affected by the various methods of governing. Further change in compression

may be obtained in several other ways provided for. The length of the connecting rod can be adjusted about $\frac{3}{4}$ inch by copper shims inserted between the rod proper and half of the end. In the second place, the length of cylinder can be slightly increased by inserting a gasket between the cylinder and frame. This same result may be obtained by the use of a piston of different length.

Two new cycles, which it is believed will materially increase the compression and the resulting mean effective pressure and power, are proposed. The theoretical indicator diagrams for these cycles and for the normal cycle are shown in Figs. 8, 9 and 10. Figs. 11 and 12 show these diagrams for the two new cycles, exaggerated in order to bring out the fundamentally new points.

The cycle with vacuum exhaust (Fig. 8.) will take place as follows: Only one exhaust valve is opened at the end of the expansion

FIG. 8
INDICATOR DIAGRAM WITH VACUUM EXHAUST

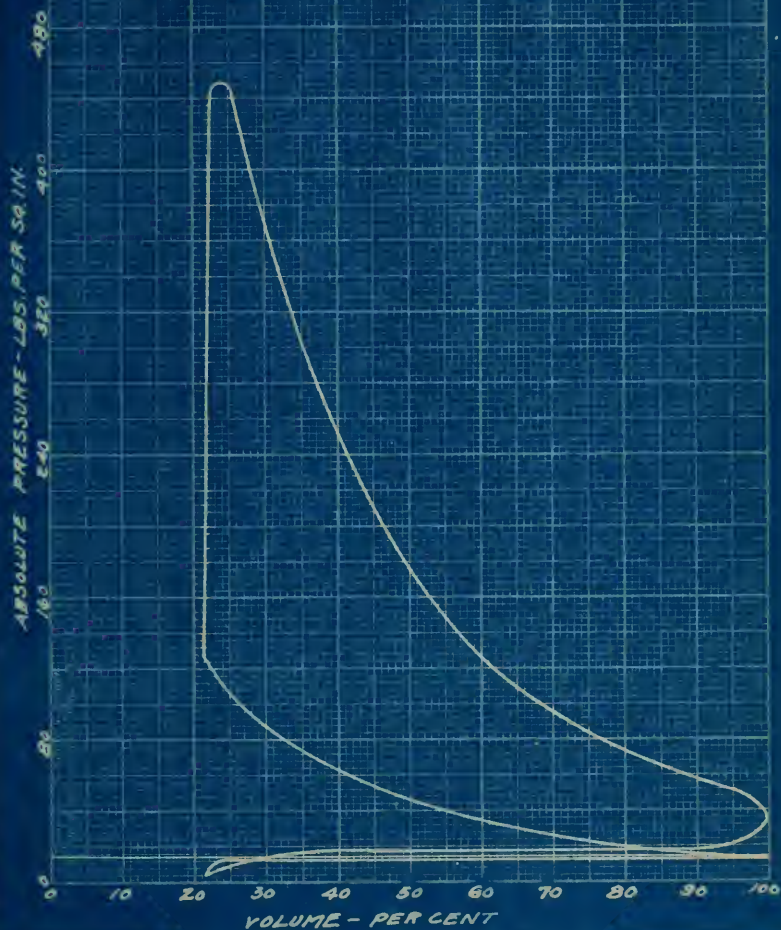


FIG. 9.
INDICATOR DIAGRAM WITH PRECOMPRESSION

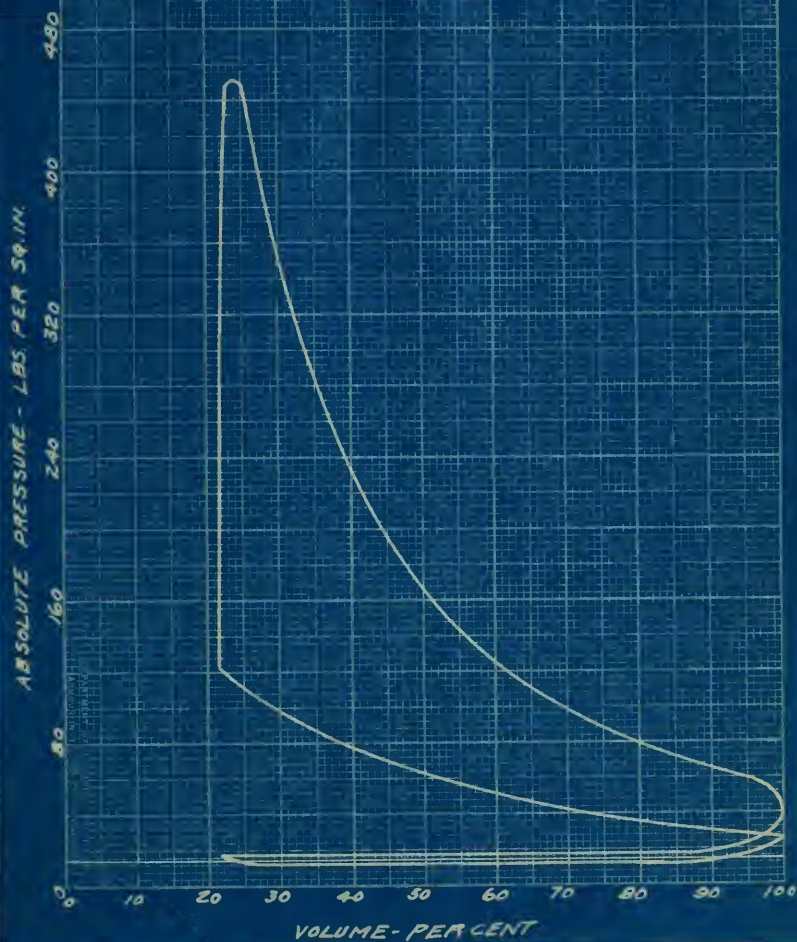
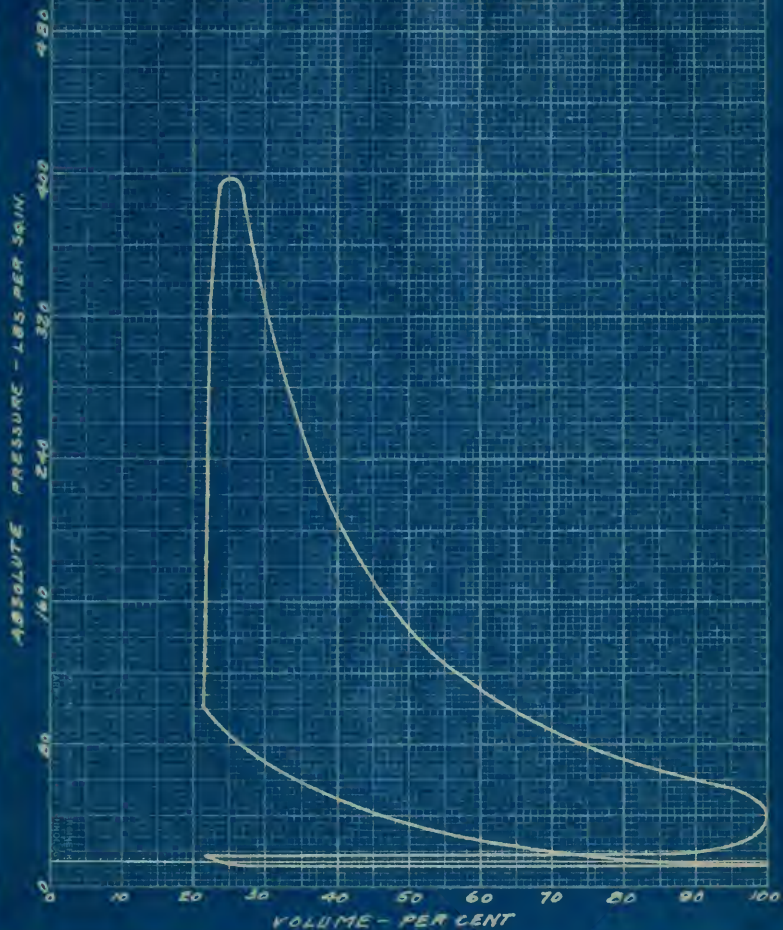


FIG. 10
NORMAL INDICATOR DIAGRAM



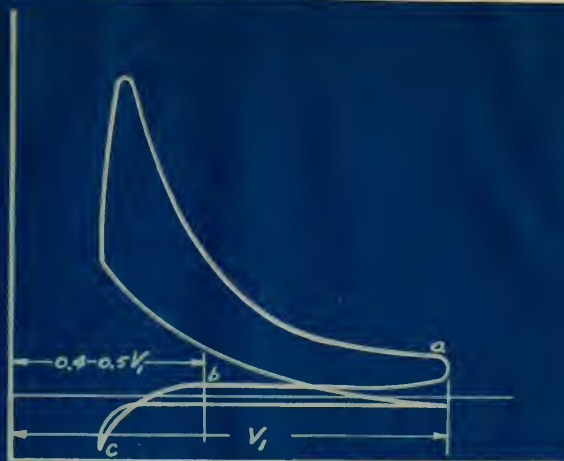


FIG. 11.
CYCLE WITH VACUUM EXHAUST.

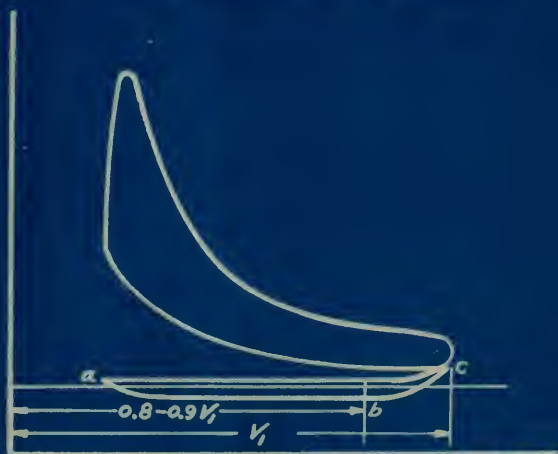


FIG. 12.
CYCLE WITH PRECOMPRESSION.

stroke at "a", and remains open to the point "b" of the exhaust stroke which is about 0.4 to 0.5 of the total cylinder volume (from head end) or approximately 0.7 of the exhaust stroke. At the point "b", or slightly before, the other exhaust valve opens; and through the connection of its port with a vacuum pump, the burned gases are exhausted from the cylinder. It is believed that the pressure at the end of the exhaust stroke can in this manner be reduced to 5 lb. per sq.in. absolute or about one-third of the normal pressure. This will then reduce the weight of exhaust gases remaining in the cylinder by two-thirds. At the end of the exhaust stroke the auxiliary exhaust valve then closes; and a fresh charge is drawn in on the inlet stroke. The higher compression and explosion pressures will be obtained by the less-diluted charge taken in.

The pre-compression cycle will necessitate the use of a new cylinder head with two

inlet valves. At the beginning of the intake stroke only one of the inlet valves is open; and it remains open to the point "b" (Fig. 9.) which is about 0.8 or 0.9 of the total cylinder volume or approximately 0.81 of the inlet stroke. At this point the first inlet valve closes, and the second or auxiliary inlet valve opens, admitting a compressed charge of gas during the remainder of the inlet stroke. A pressure of 20 to 25 lb. per sq.in. abs. at the end of the inlet stroke can thus be obtained; and higher compression and explosion pressures will result.

CONCLUSION.

Conclusion. (Future Considerations).

1. Valve Gear.

The part of the valve gear attached directly to the cylinder head is somewhat bulky ; and with further study and experiment could probably be simplified and perfected.

2. Flywheel and Crankshaft.

The rational design and theoretical analysis of the flywheel and crankshaft were beyond the scope of the present work. These would form an important part of the work in the further study of the design of this engine. Included in this study will be the various inertia and shaking force diagrams to show the necessary balancing required.

3. Bearings.

A study of the bearings required and a determination of the most suitable type for each location must be made.

4. Lubrication.

THE HISTORY OF THE

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THE HISTORY OF THE

Pressure lubrication with continuous circulation has been decided upon. The best means of lubricating the crank pin and piston pin must be considered.

5. Governing.

The governing mechanism will be studied and designed more in detail.

6. Ignition.

Magneto ignition has been decided upon; but the method of spark control will be taken up.

7. Carburetion.

Carburetion will have to be studied in conjunction with governing. It is recommended that the Stewart, vacuum-feed system be considered. A tank can be placed at the bottom of the frame.

APPENDIX.

Appendix.Bibliography.Cylinder:

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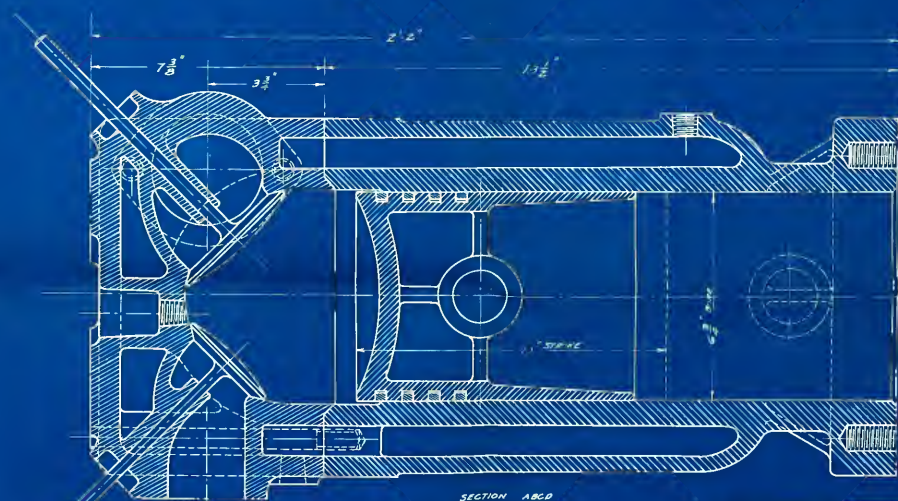
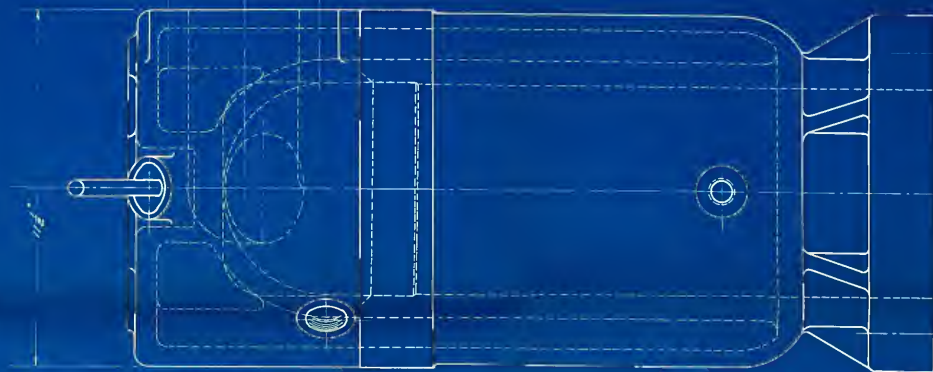
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SECTION ABCD

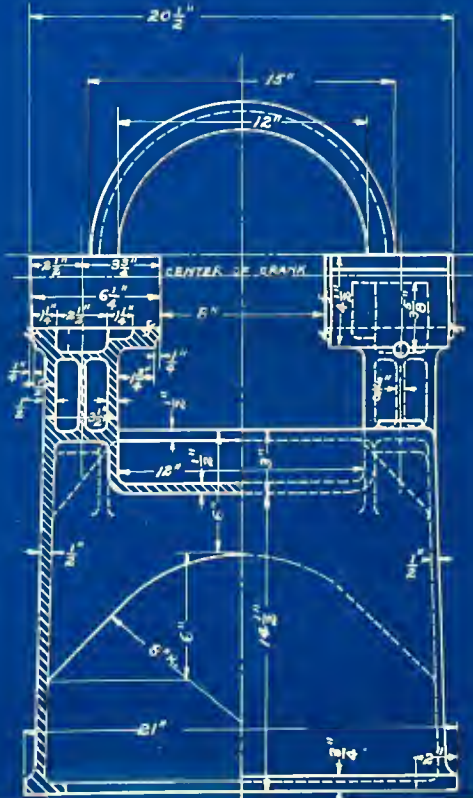
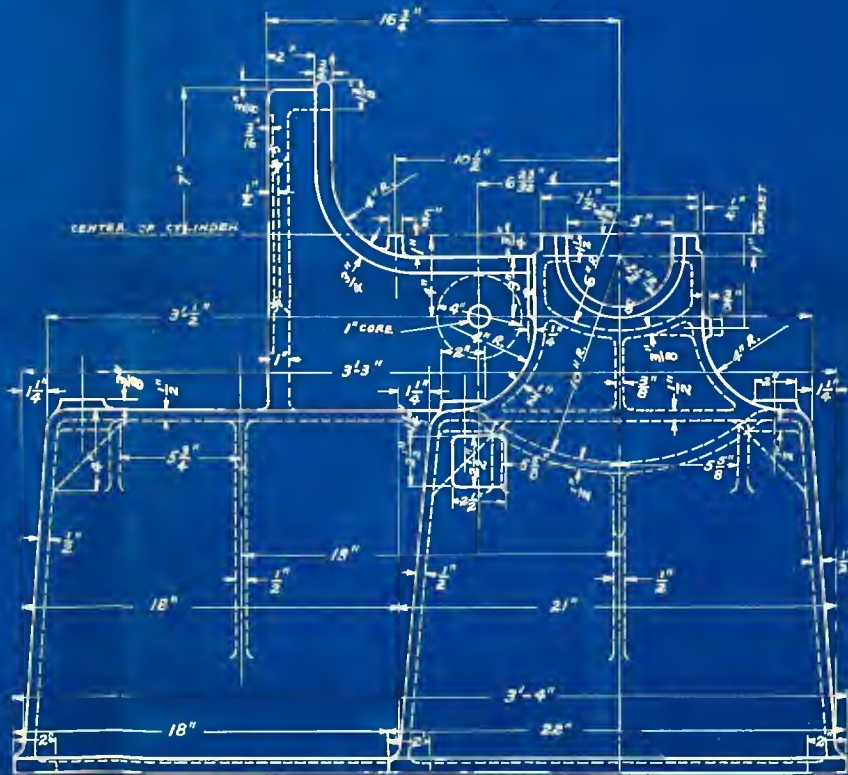
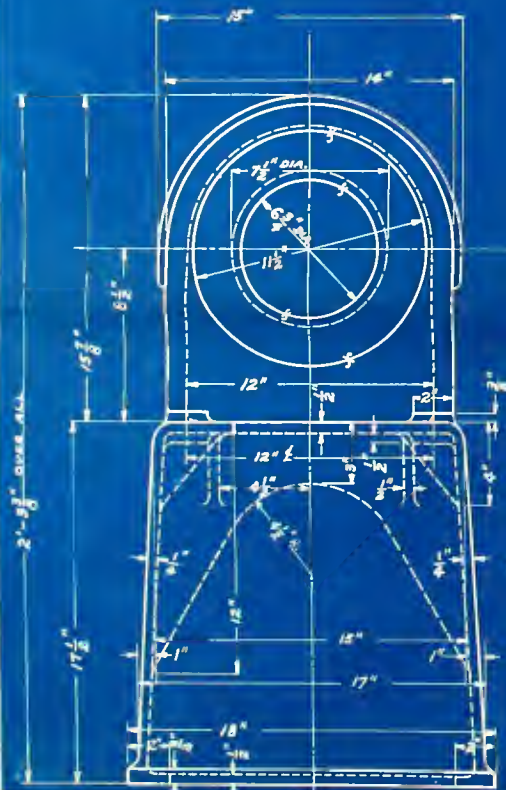
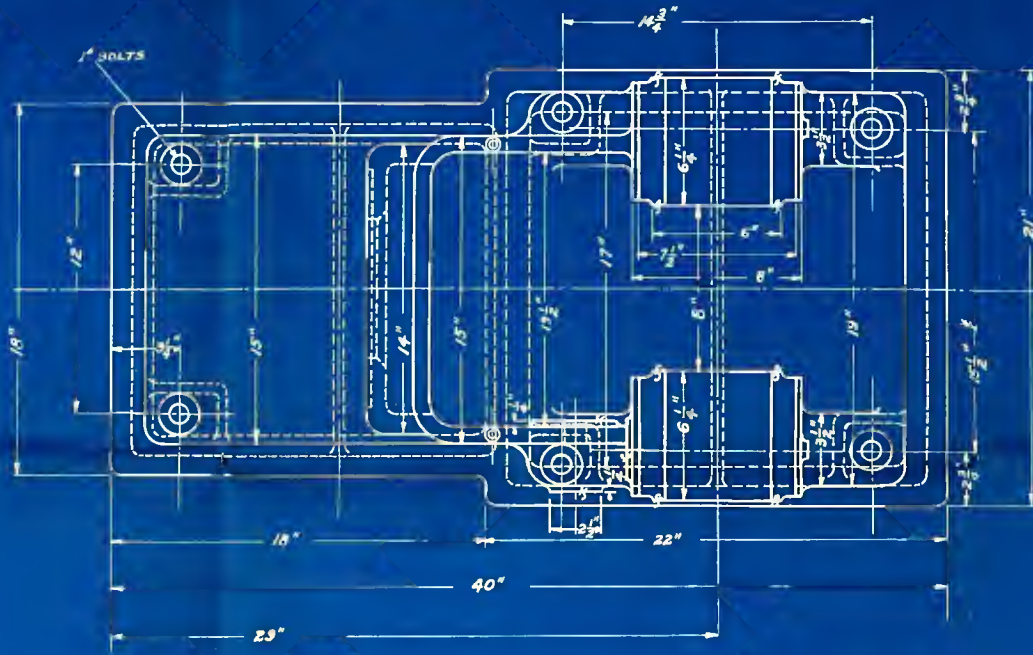
DESIGN OF A 10 H.P. GAS ENGINE
CYLINDER LAYOUT

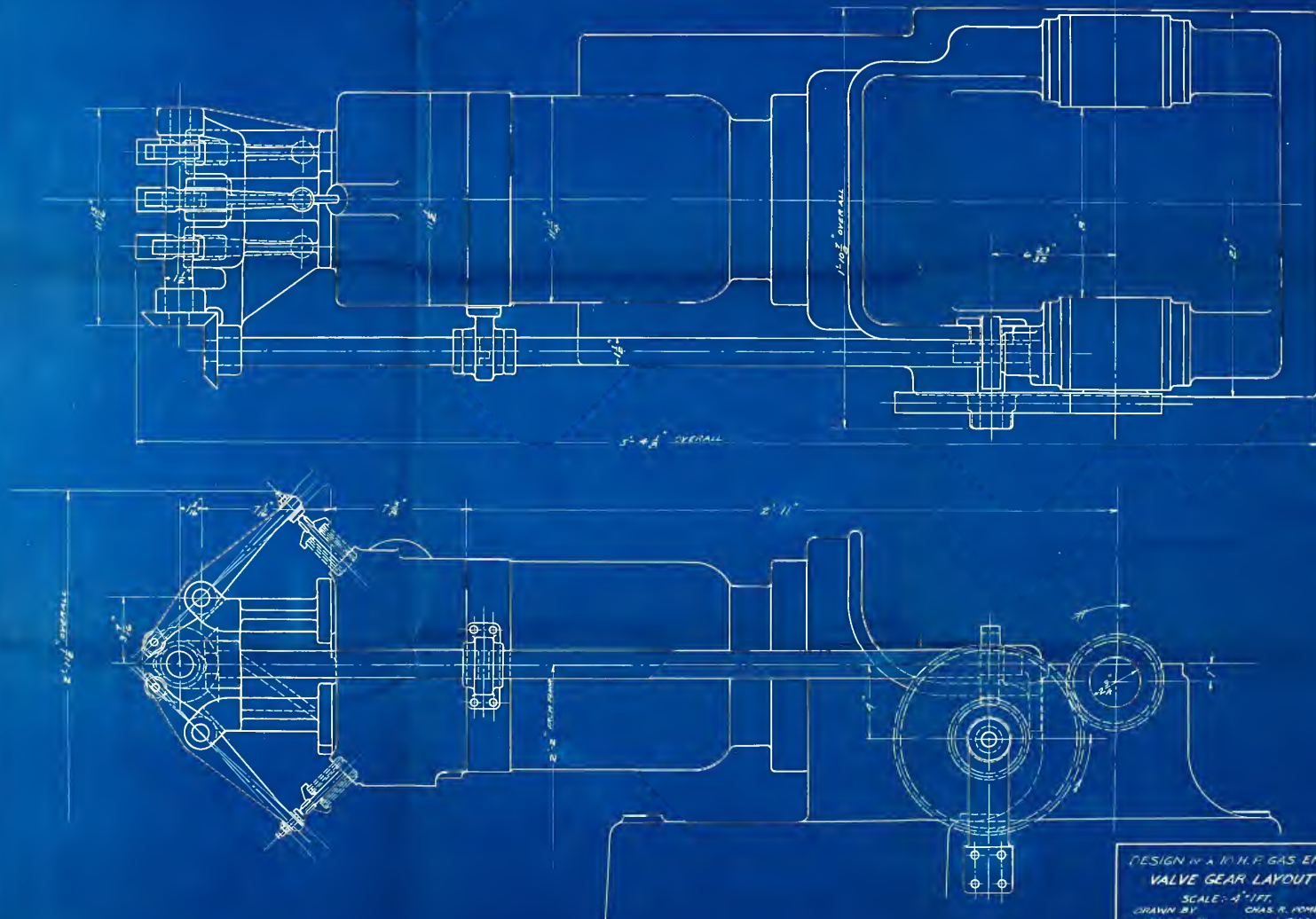
SCALE: 6"=1 FT.

DRAWN BY CHAS. R. JONES
CHECKED BY AM. J. JONES
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.
THESIS 1917 PLATE I.

DESIGN OF A 10 H.P. GAS ENGINE
DETAIL OF FRAME

SCALE: 3" = 1 FT.
DRAWN BY A. J. PLOCHINSKY
CHECKED BY C. R. POMEROY
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.
THESIS 1917. PLATE 2.





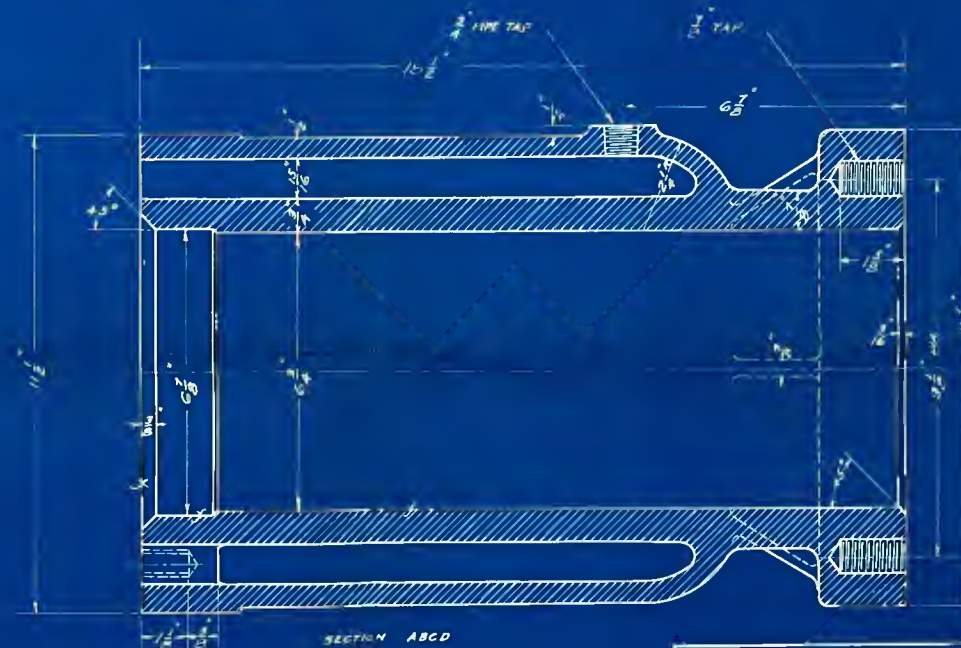
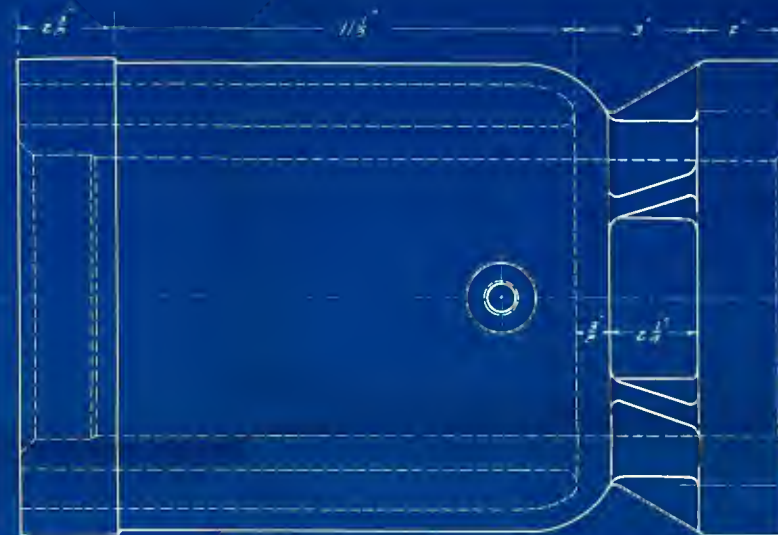
DESIGN OF A 10 H.P. GAS ENGINE
VALVE GEAR LAYOUT

SCALE: 1/4" = 1"

DRAWN BY: CHAS. R. POWERIT,
ENGINEER BY: ARTHUR W. POWERIT,
ARMOUR INSTITUTE OF TECHNOLOGY,
CHICAGO, ILL.

THESES 1817

PLATE 3.



DESIGN OF A 10 H.P. GAS ENGINE
CYLINDER DETAILS

SCALE: 6" = 1 FT.
DRAWN BY CHAS. R. POWEROY
CHECKED BY ABE J. FLODINSKY
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.
THESIS 1917 PLATE 4



SPUR GEAR
STEEL ONE REQD.
60 D.P. PITCH 40 TEETH



JACK SHAFT
STEEL ONE REQD.



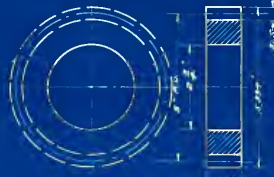
PART OF LAY SHAFT
STEEL ONE REQD.



PART OF LAY SHAFT
STEEL ONE REQD.



CAM SHAFT
STEEL ONE REQD.



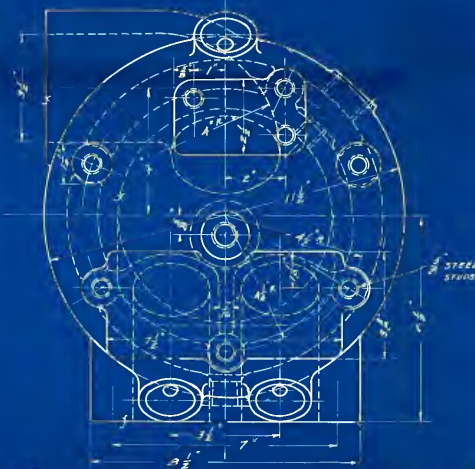
SPUR GEAR
STEEL ONE REQD.
60 D.P. PITCH 20 TEETH



SPUR GEAR
STEEL ONE REQD.
60 D.P. PITCH 20 TEETH



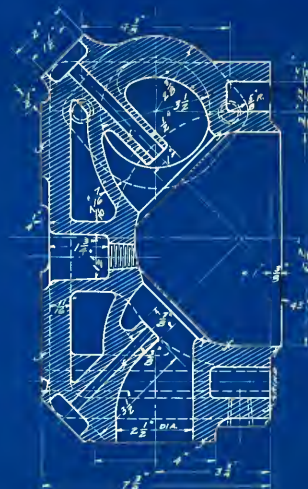
SPUR GEAR
STEEL ONE REQD.
60 D.P. PITCH 20 TEETH



CRANKSHAFT
CAST IRON ONE REQD.
(FOR PLAN VIEW SEE PLATE 1)



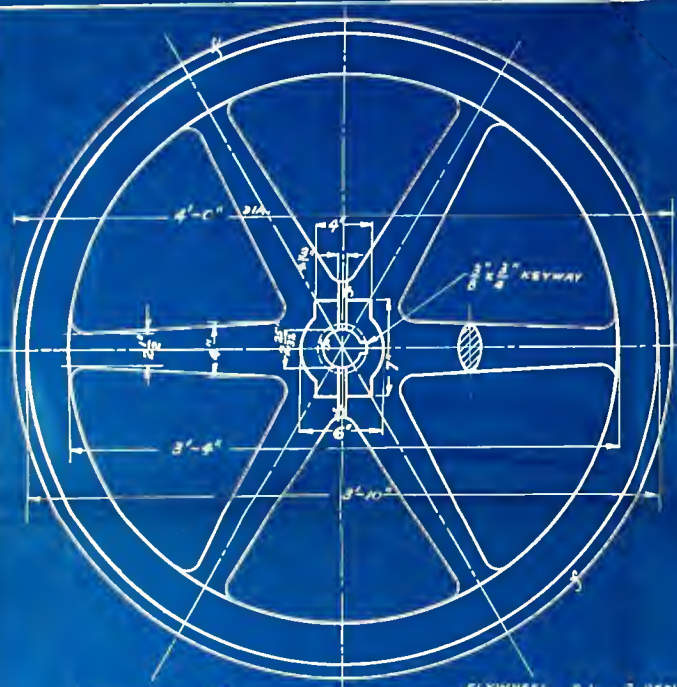
ROCKER ARM
STEEL THREE REQD.



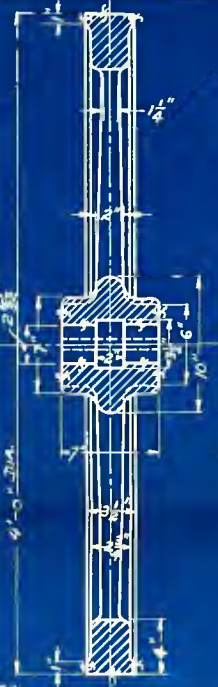
ROCKER
TWO STEEL THREE REQD.

DESIGN OF A 10 H.P. GAS ENGINE VALVE GEAR DETAILS

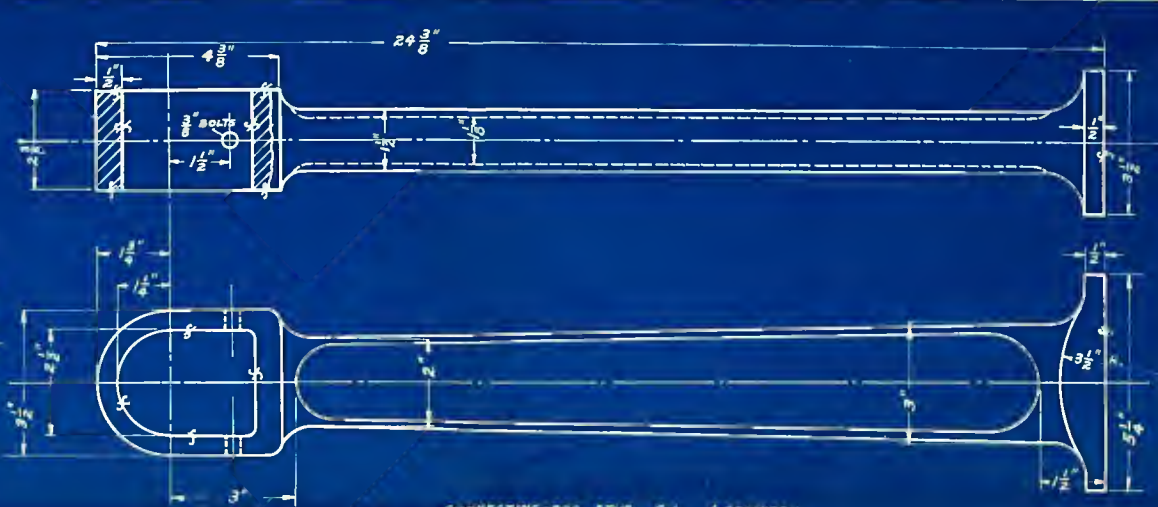
SCALE: 6"=1 FT.
DRAWN BY CHAS. R. DIMICK
CHECKED BY ANNE J. DIMICK
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.
THESE 1917 PLATE 5



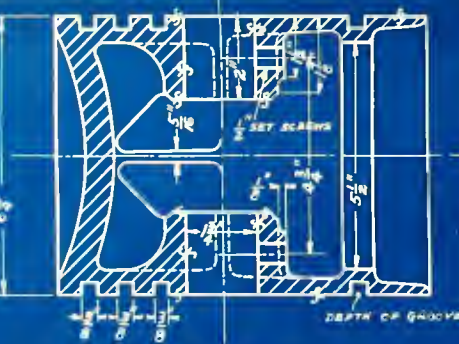
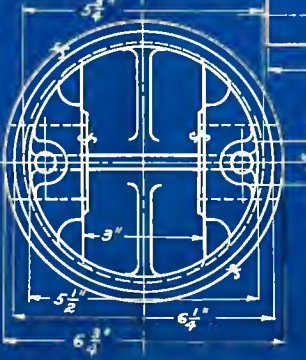
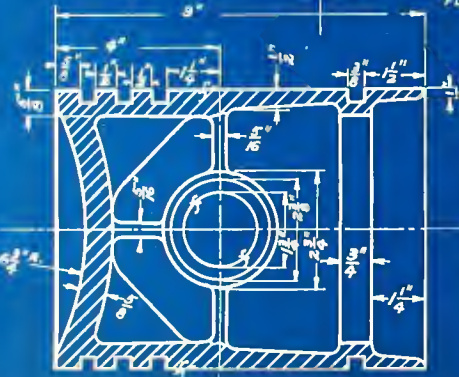
FLYWHEEL C.I. 2 REQUIRED.
SCALE: 2"=1 FT.



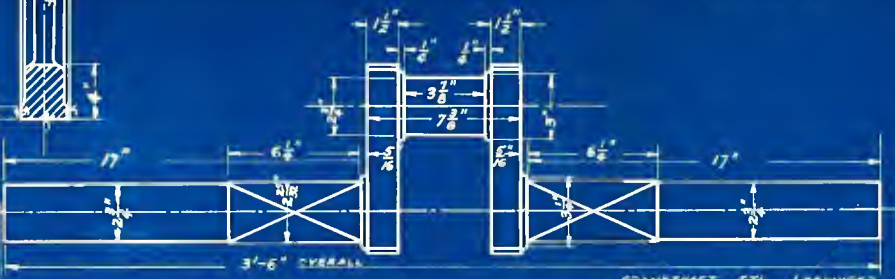
PISTON PIN STL. 1 REQUIRED.
SCALE: 6"=1 FT. F.A.D.



CONNECTING-ROD STUB. C.I. 1 REQUIRED.
SCALE: 6"=1 FT.



PISTON C.I. 1 REQUIRED.
SCALE: 6"=1 FT.



CRANKSHAFT STL. 1 REQUIRED. F.A.D.
SCALE: 3"=1 FT.



LARGE END CONNECTING-ROD (SECT 2).
C.I. 1 REQUIRED. SCALE: 6"=1 FT. F.A.D.



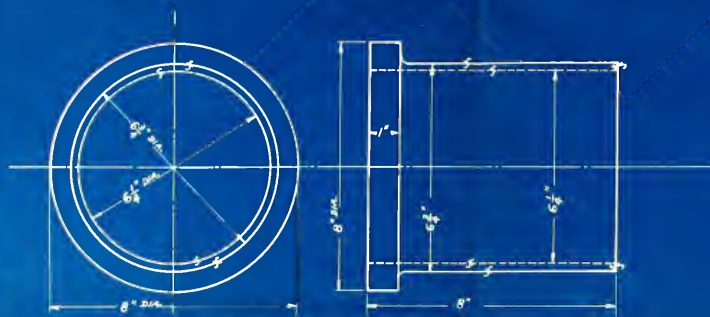
LARGE END CONNECTING-ROD (SECT 3).
C.I. 1 REQUIRED. SCALE: 6"=1 FT. F.A.D.



BRONZE BUSHING. 1 REQD.
SCALE: 6"=1 FT. F.A.D.

BRONZE BUSHING. 1 REQUIRED.
SCALE: 6"=1 FT. F.A.D.

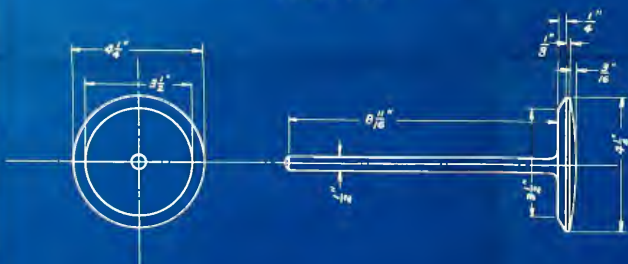
**DESIGN OF A 10 H.P. GAS ENGINE
RECIPROCATING PARTS**
SCALE: AS INDICATED.
DRAWN BY A. J. FLODINSKY.
CHECKED BY C. R. POMEROY.
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.
THESIS 1917. PLATE 6.



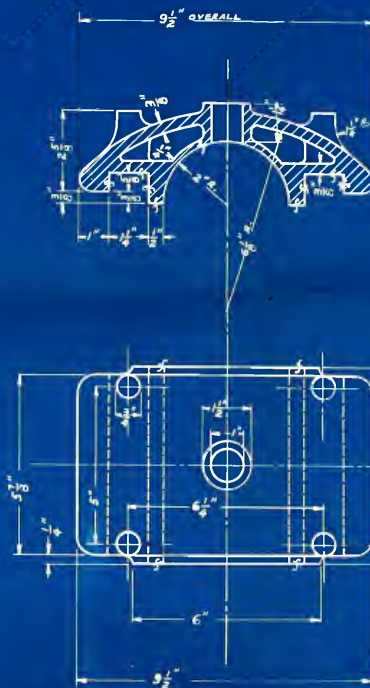
PST PATTERN FOR CAST-IRON PISTON RINGS
SCALE: 6"=1 FT.



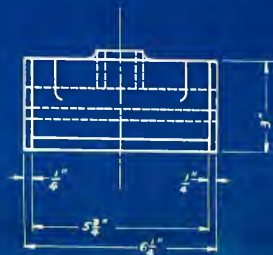
EXHAUST VALVE TUNGSTEN STL. 2 REQD.
SCALE: 6"=1 FT.



INLET VALVE TUNGSTEN STL. 1 REQD.
SCALE: 6"=1 FT.



BEARING CAP C.I. 2 REQUIRED.
SCALE: 6"=1 FT.



DESIGN OF A 10 H.P. GAS ENGINE
MISCELLANEOUS PARTS

SCALE: 6"=1 FT.

DRAWN BY A.J. MLOCHINSKY.

CHECKED BY C.R. POMEROY.

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THESIS 1917. PLATE 7.

